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INVESTIGATION OF A LOW-COST SERVOACTUATOR FOR HYSAS

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APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

This report has been reviewed by the Applied Technology Laboratory, U.S. Army Research and Technology Laboratories (AVRADCOM), and is considered to be technically sound.

This research effort resulted from the need which exists for a low-cost, simple fluidic stability augmentation system (SAS) servoactuator with sufficient performance to meet essential SAS operational requirements, but without the high cost performance features, which are not needed, associated with the currently used servoactuators. Gain and null variations over the desired fluid operating temperature range, continue to be the only significant limitations to the servoactuator performance.

Mr. George W. Fosdick of the Systems Support Division served as the project engineer for this effort.

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This document covers the design and testing of a fluidic input servoactuator to perform the series servoactuator function in a helicopter hydrofluidic stability augmentation system (HYSAS). The servoactuator consists of a two-stage fluidic amplifier cascade driving a conventional spool valve that positions a spring-centered cylinder. Simplicity and minimum cost commensurate with essential servoactuator performance was the design goal. A

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20. Abstract (Cont.)

breadboard model servoactuator was designed, fabricated, and bench tested to evaluate concept feasibility. Servoactuator performance objectives were met at nominal supply conditions, but not over the complete operational oil temperature range.

PREFACE

This document is the final report on a program to investigate the feasibility of a simple fluidic input servoactuator to perform the series servoactuator function in a helicopter hydrofluidic stability augmentation system (HYSAS). The work covered in this report was performed from April 1977 to March 1978 by the Avionics Division of Honeywell Inc., under Contract DAAJ02-77-C-0025. The sponsoring agency was the Applied Technology Laboratory of the U. S. Army Research and Technology Laboratories, Fort Eustis, Virginia, with Mr. George Fosdick as Project Engineer.

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INTRODUCTION

The extensible link or series servoactuator, which has been used with fluidic stability augmentation systems (SAS) for helicopters, is a modification of a conventional electrohydraulic servoactuator. The electromagnetic torquer (coil) is replaced with force capsules (bellows) to provide an input capability compatible with the fluidic SAS controller output, and mechanical feedback from the output cylinder to the armature is retained. All other servoactuator characteristics are similar to those of an electrohydraulic servoactuator.

Although this type of servoactuator has been in operation for some time and significant numbers have been built (more than 50), servoactuator cost remains high. Some of the reasons for this high cost are the relatively high level of performance specified, the specialized configuration, and the relatively low production volume as compared to electrohydraulic units. At present, the fluidic input servoactuator accounts for between 50 and 60 percent of the cost of the fluidic SAS.

One of the competitors of the fluidic SAS is an electrical SAS controller (conventional rate gyro and electronics) driving an electromechanical servoactuator (motor and gears). This system has a cost comparable to or even lower than the fluidic SAS. One of the primary reasons for this is the lower cost of the electromechanical actuator. In certain helicopter SAS applications, the reduced performance of the electromechanical servoactuator has been acceptable in view of this lower cost.

The purpose of this program is the development of a low-cost fluidic input servoactuator with sufficient performance to meet essential SAS operational requirements. Past programs to develop fluidic servoactuators have been relatively unsuccessful in servoactuator cost reduction, primarily due to the fact that the performance and features of the present higher cost servoactuators have been used as design requirements. Therefore, the problem solution becomes one of developing a fluidic servoactuator concept with performance and features equivalent to the essential requirements for a helicopter fluidic SAS servoactuator. A summary of past fluidic servoactuator programs is presented in Appendix B.

The objective of this program is to investigate the feasibility of a simple fluidic amplifier-driven servovalve driving a spring-centered cylinder to perform the series servoactuator function in a helicopter hydrofluidic stability augmentation system (HYSAS). This report covers the design and testing of a breadboard model servoactuator to evaluate concept feasibility.

SECTION II
SERVOACTUATOR DESIGN

The initial servoactuator concept consisted of a high-pressure fluidic amplifier cascade driving a spring-centered cylinder. During the initial design of this configuration and after preliminary fluidic amplifier tests had been run, it became apparent that the proposed concept had significant performance limitations in the areas of response and flow consumption. As a result, the servoactuator design approach was modified to include a conventional spool valve between the fluidic amplifier and the cylinder. This section presents the performance objectives for the servoactuator and a description of both the initial design approach, which was subsequently discontinued, and the alternate design approach, which was tested on this program.

PERFORMANCE OBJECTIVES

The principal performance objectives for the servoactuator are given below and a servoactuator specification is presented in Appendix A of this report.

Output Stroke: ± 0.200 in.

Output Force (at any stroke position): Minimum - 2.0 lb.
Maximum - to be consistent with the design, objectives of low cost, lightweight, etc.

Supply Flow: 1.0 in.^{3}/sec (max)

Fluid/Temperature: MIL-H-5606C at 100° ± 10°F

External Load Resistance: Irreversible within ± 5% of stroke (at any position) when subjected to an external load greater than the actuator output force.

Dynamic Response (no load): Equivalent to a -3 dB bandwidth of 3 Hz

Controller Compatibility: Capability of operating with fluid signals provided by existing HYSAS controllers.

INITIAL SERVOACTUATOR CONCEPT

Concept Selection

Figure 1 shows a typical power jack and SAS actuator installation. With a pilot input that produces a movement of the control linkage (X_i), the control valve spool is deflected causing an oil flow to the

power jack piston. The power jack piston moves, causing the control valve to recenter so that the power jack output (X_o) equals the control linkage motion (X_i). When the HYSAS detects an angular rate, the HYSAS actuator provides an actuator output (ΔX) to the control valve spool that is superimposed on the pilot commanded input. The HYSAS actuator acts as an extensible link in the control linkage so that the power jack output equals the combined pilot and SAS input ($X_i \pm \Delta X = X_o$).

The HYSAS servoactuator consists of a fluidic amplifier cascade, a spring-centered cylinder, and a pressure-off cylinder lock. Figure 2 is a functional block diagram of the servoactuator concept.

In operation, the application of supply pressure pressurizes the fluidic amplifier cascade and mechanically unlocks the power cylinder. A control pressure signal from the fluidic controller (ΔP_c) causes the fluidic amplifier cascade to produce an output differential pressure/flow. The power cylinder strokes until the pressure-generated cylinder force is balanced by the cylinder centering spring force. The servoactuator output position is proportional to the input control signal pressure. High rate centering springs are used in the power cylinder to minimize the effects of load force on the servoactuator scale factor. The high rate centering springs also are used to provide servoactuator stiffness (cylinder resistance to changing external load). To provide 100 percent irreversibility in the absence of hydraulic supply pressure, a mechanical locking device locks up the power cylinder, and the servoactuator acts as a rigid link in the control linkage.

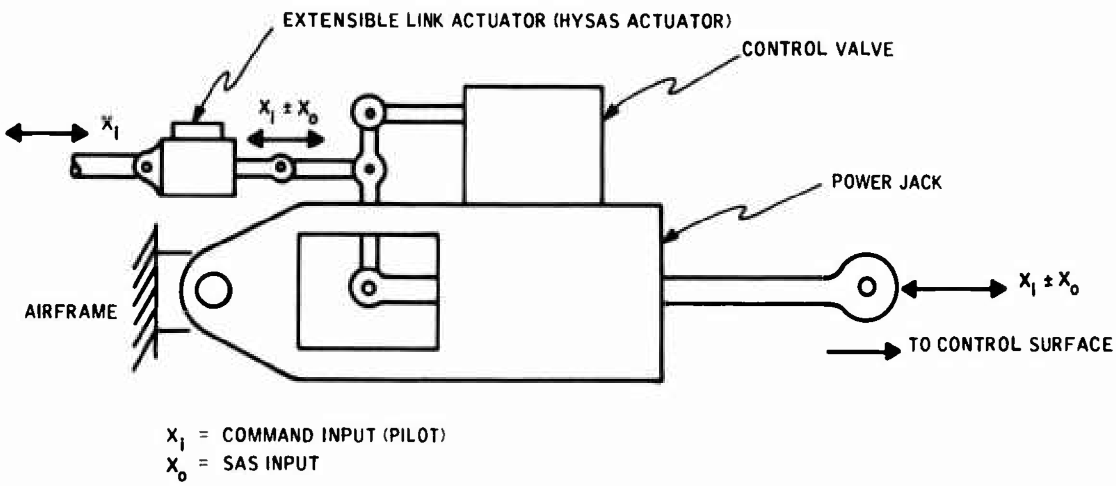


Figure 1. Typical Power Jack and SAS Actuator Installation

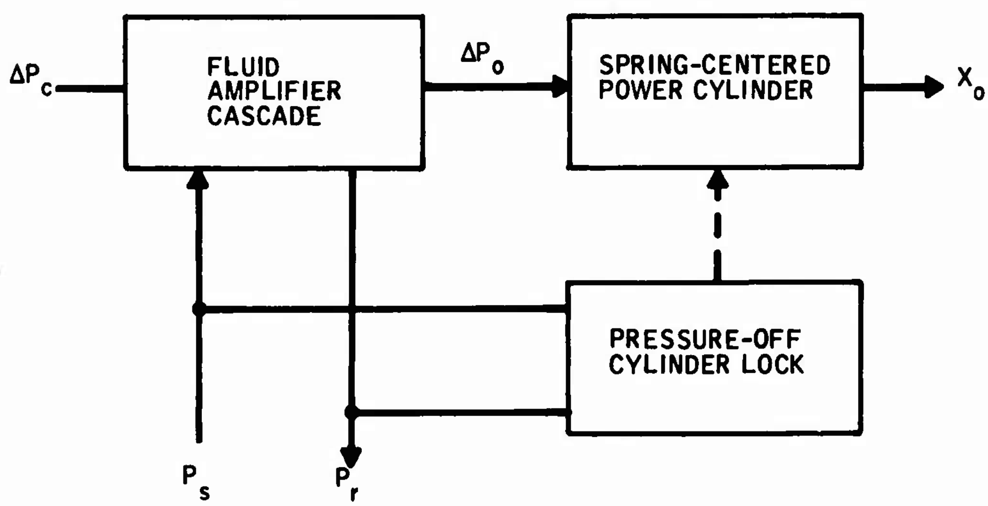


Figure 2. Servoactuator Block Diagram (Initial Concept)

The servoactuator no load transfer function is approximated by the expression

$$G(S) = \frac{X_o}{\Delta P_c}(S) = \frac{K_a A_p}{K_s}\left[\frac{1}{\left(\frac{A_p^{\,2}}{K_s}\right)\left(\frac{\Delta P_o}{\Delta Q_o}\right) S + 1}\right] \text{ in./psid}$$

Where:

K_a = fluid amplifier cascade pressure gain $\left(\frac{\text{psid}}{\text{psid}}\right)$

A_p = power cylinder area (in.^{2})

K_s = centering spring effective spring rate $\left(\frac{\text{lb.}}{\text{in.}}\right)$

$\frac{\Delta P_o}{\Delta Q_o}$ = power amplifier stage output resistance $\left(\frac{\text{psi}}{\text{in.}^3/\text{sec}}\right)$ (average slope of P-Q curve)

ΔP_c = input signal differential pressure (psid)

X_o = servoactuator output position (inches)

External Load Resistance Discussion

The actuator external load resistance or stiffness is a measure of the servoactuator's ability to hold the command position under changing load conditions. Because the HYSAS servoactuator is in series with the pilot's command linkage to the boost jack control valve, the servoactuator's operating stiffness must be adequate to transmit the

command link motion with sufficient fidelity. To obtain 100 percent transmission fidelity, the servoactuator would have to have 100 percent irreversibility. Inspection of the servoactuator transfer function shows that the simple spring-centered servoactuator stiffness is a function of the spring rate of the centering spring and the ratio of the pressure-generated cylinder force to the deflection-generated centering spring force. Therefore, any external load forces in excess of the net cylinder force generated by the fluidic amplifier cascade output pressure would cause the cylinder to back up.

The recommended stiffness objective of ±10 percent stroke with the application of ±2.0 pounds external load was based on the simple open-loop servoactuator concept. The 2.0-pound output force was based on a value of four times that stated as normal for typical boost jack control valve operation. If the boost jack is operating normally, the normal deviation in control linkage transmission fidelity caused by the stiffness recommendation would be approximately 0.25 percent of the pedal motion. If the boost jack control valve force increased for some reason to 2.0 pounds, the deviation in control linkage transmission fidelity would be approximately 1.0 percent of the pedal motion. If the boost jack control valve jammed solid, the deviation in control linkage transmission fidelity would be approximately 10 percent of the pedal motion (0.200 inch actuator motion equals 10 percent pedal motion). If the HYSAS supply pressure is turned off, the servoactuator would be mechanically locked and the control linkage transmission fidelity becomes 100 percent.

Servoactuator Sizing Calculations

Servoactuator sizing is based on performance objectives. Inspection of the servoactuator transfer function showed:

- Servoactuator output force to the load is a function of cylinder area, fluid amplifier cascade output pressure, and power cylinder centering-spring force. Cascade power amplifier output pressure is a function of supply pressure.

- Servoactuator scale factor is a function of fluid amplifier cascade pressure gain, cylinder area, and spring rate of the cylinder centering springs.

- Servoactuator dynamic response is a function of the first-order lag time constant determined by the power fluid amplifier output resistance and the spring-centered power cylinder capacitance. These parameters are related to the cylinder area, the spring rate of the centering springs, and the power amplifier P-Q curve slopes.

- Although not indicated from the servoactuator transfer function, the fluid amplified cascade supply flow is a function of supply pressure, fluid amplifier power nozzle size, and number of amplifiers in the cascade.

From the preceding discussion, it is seen that there is a significant interdependance of design parameters on the various performance parameters. To initiate servoactuator sizing, it is necessary to select some parameter and initial conditions for calculation purposes.

Iterative calculations will then determine potential conflict between performance objectives. As a starting point, the design goal performance objectives are ranked in descending order of importance for design calculation purposes.

1. Output force to load and stiffness
2. Dynamic response
3. Supply flow rate
4. Servoactuator scale factor

With respect to the performance objective order of ranking, output force and stiffness are considered to be critical. The actuator must have sufficient force to actuate the boost jack control valve; and the actuator stiffness, when the actuator is operating, must be sufficient to permit reasonable control linkage motion transmission fidelity. A servoactuator used in a HYSAS application must have these capabilities. Minimum dynamic response is considered next in importance,as dynamic response must be adequate to meet minimum HYSAS system requirements. Supply flow is considered next,as the helicopter HYSAS hydraulic power supply capacity is limited. Servoactuator scale factor is considered last,as this is a factor in overall HYSAS system gain, and it is understood that the gains of existing HYSAS controllers could be easily increased, if necessary.

Power Cylinder - Power cylinder sizing is based on the following assumptions:

- Pressure-generated cylinder force (F_p) at maximum input to the fluid amplifier cascade is 22 pounds.

- Spring force (F_s) at maximum cylinder stroke is 20 pounds to provide 2 pounds force to load at maximum cylinder stroke and a maximum input signal.

- Fluid amplifier cascade output pressure (ΔP_o) at maximum input signal is 110 psid.

$$F_p = \Delta P_o A_p = 110 A_p = 22 \text{ lb}$$

$$A_p = 22/110 = 0.20 \text{ in.}^2$$

$$F_s = K_s X_o = 20 \text{ lb}$$

$$X_o = \pm 0.200 \text{ in.}$$

$$K_s = 20/0.20 = 100 \text{ lb/in.}$$

For two springs initially in compression, the individual spring rate is one half the effective composite spring.

$$K'_s = \frac{K_s}{2} = 100/2 = 50 \text{ lb/in.}$$

The power cylinder design parameters are:

- $\pm$ 0.200 in. stroke
- Effective area = 0.20 in.^{2}
- Spring rate of each centering spring = 50 lb/in.

With the power cylinder centering springs selected to provide an effective spring rate of 100 lb/in. (K_s), the actuator stiffness is 100 lb/in. The actuator output force to the load at maximum input signal will be:

- at mid-position the output force is 22 lb
- at one-half stroke the output force is 12 lb
- at full stroke the output force is 2 lb.

Fluid Amplifier Cascade - The fluid amplifier cascade design is based on the following assumptions:

- Due to the limited scope of the program, an existing fluid amplifier design was used, as the program did not allow for special amplifier development work.
- The AC 17920-43 fluid amplifier design was selected. This is a power-type fluid amplifier with a 0.020-X 0.020-inch power nozzle developed for actuator driving applications using supply pressures in the 150 to 2000 psig range.

A two-stage AC 17920-43 fluid amplifier cascade was tested at supply pressures of 140 psid, 200 psid, and 400 psid over an oil temperature range from 83° to 175°F. Figure 3 shows cascade blocked actuator port pressure gain curves for three supply pressures at the nominal test oil temperature condition. Figure 4 shows a plot of pressure gain versus oil temperature. It is seen that relatively constant pressure gain is achieved at the 400 psid supply condition. Reynolds number approximations indicate that the cascade is operating in the turbulent flow regime at the 400 psid supply condition. At the 140

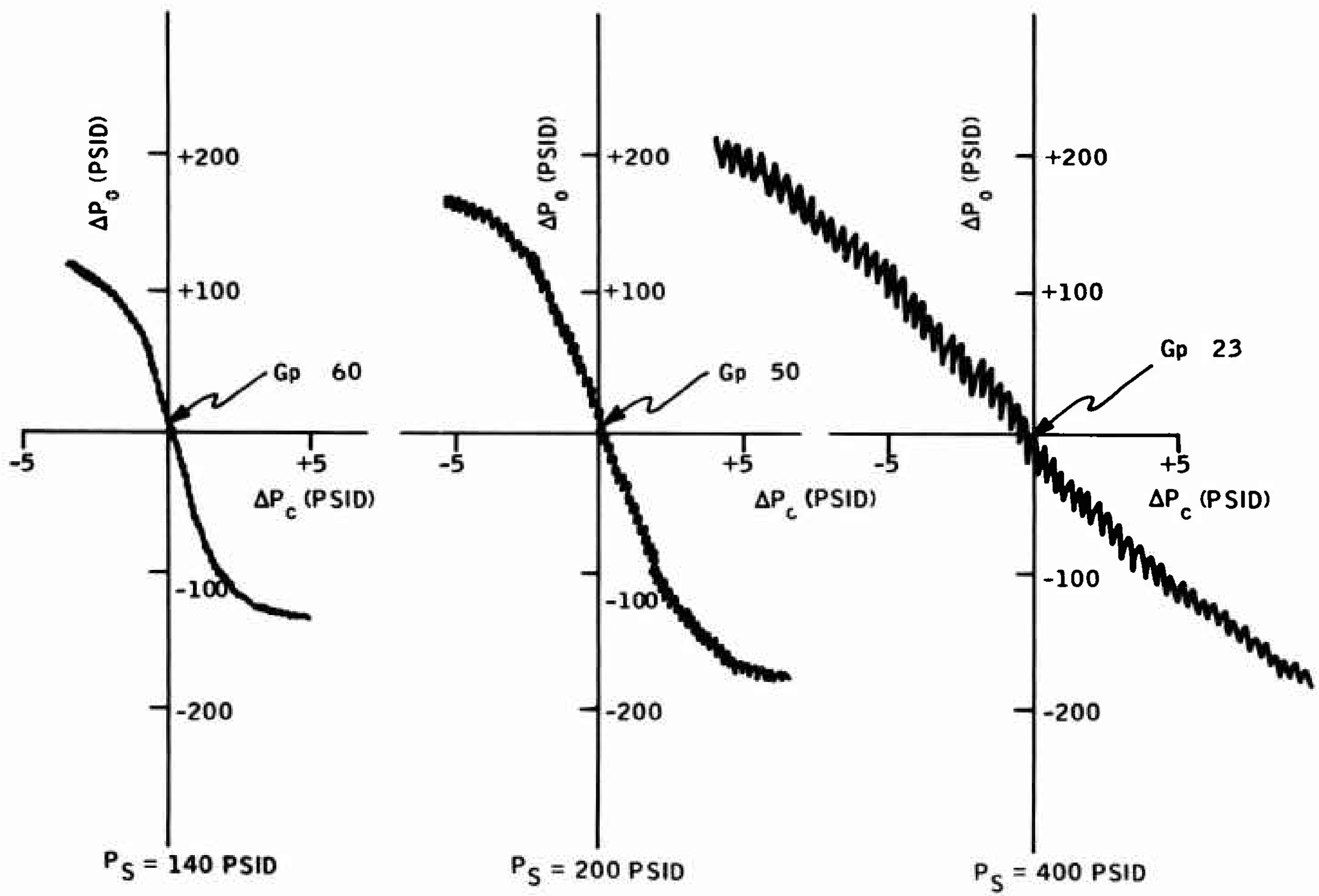


Figure 3. Typical Pressure Gain Curves. AC 17920-43 Two-Stage Cascade

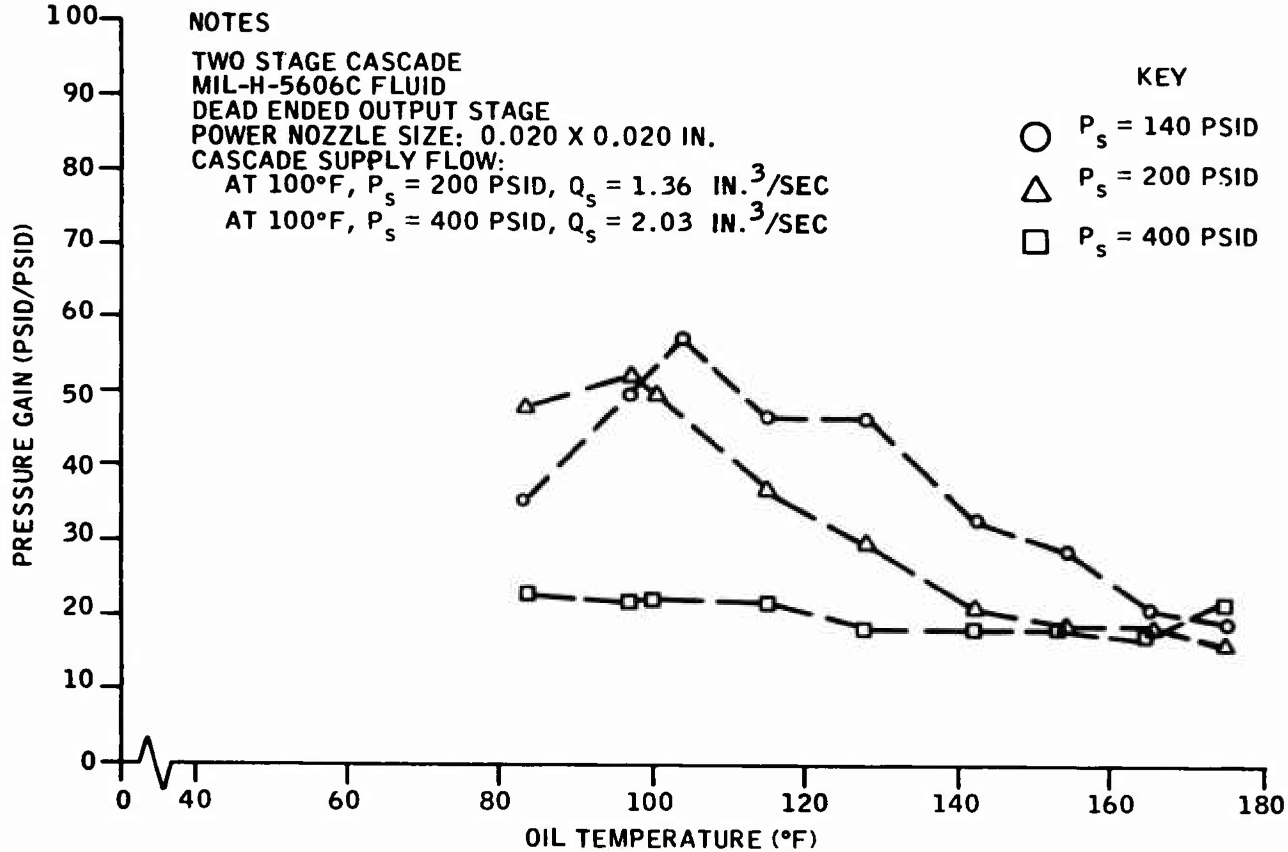


Figure 4. Pressure Gain versus Oil Temperature Plot

psid and 200 psid supply conditions, the cascade operates over the laminar, transition, and turbulent flow regimes. The cascade gain change with oil temperature change is believed to be due to changes in power jet velocity profile, jet deflection sensitivity, and other factors. To operate over the design goal oil temperature range with relatively constant gain, it is concluded that the cascade must be operated in the turbulent flow regime. For the 400 psid supply pressure condition, it is estimated that the cascade gain will increase somewhat as the oil temperature is reduced below 80°F.

Figure 5 shows the measured load pressure-load flow (cylinder ports in series) (P-Q) envelope limit curves for the AC 17920-43 amplifier at the 140 psid, 200 psid, and 400 psid supply pressures. Available test equipment was not adequate for P-Q curve measurements at lesser input signal pressures. Normally, the P-Q curve shapes at lesser input signals are the same as the limiting envelope, but terminate at lesser load flow and load pressure levels. The servo-dynamic response bandwidth is a function of the power amplifier output resistance, as defined by the slope of the P-Q curve. For the cylinder design parameters defined, a P-Q curve slope (average) of 130 psi/cis is necessary to achieve the -3 dB bandwidth of 3 Hz. From the measured data, it was concluded that the output resistance of the AC17920-43 amplifier design is too great for the intended application.

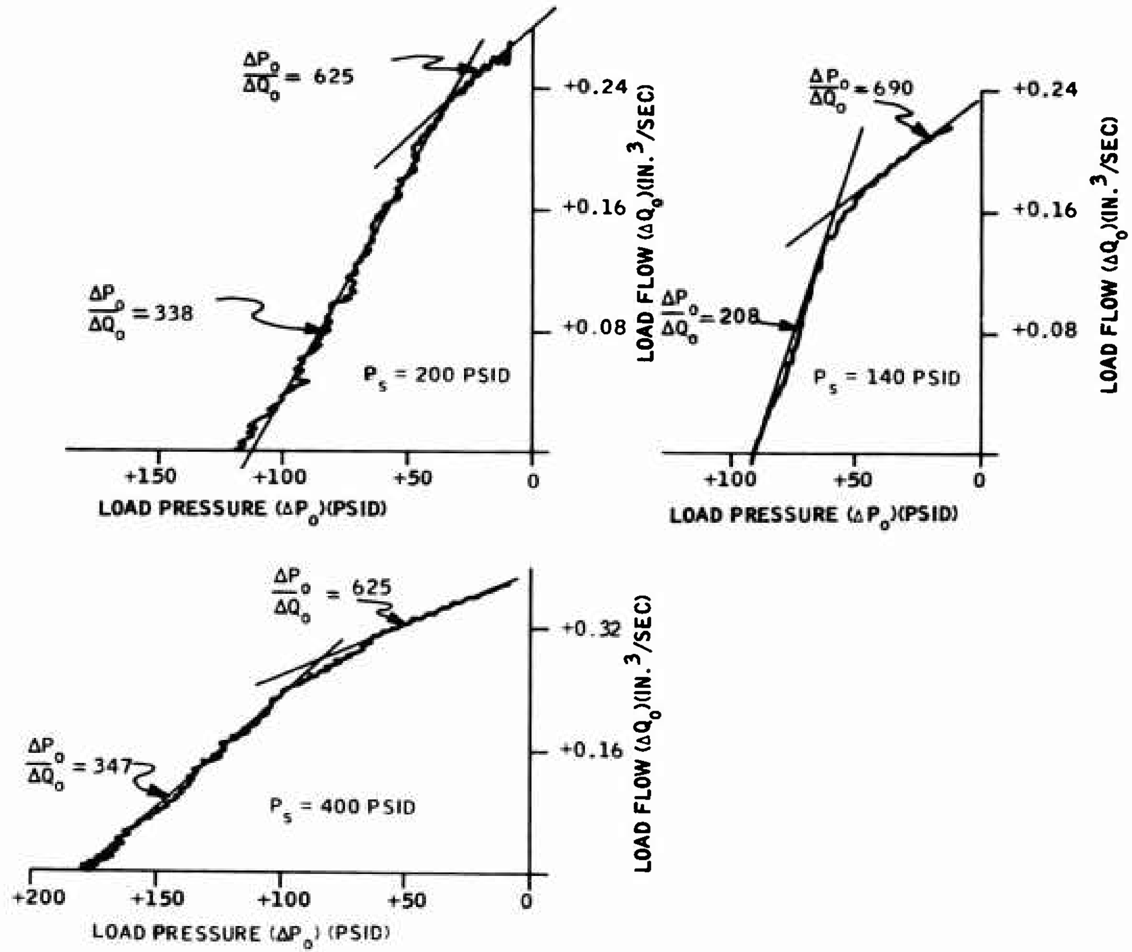


Figure 5. Load Pressure -- Load Flow Curves

Servoactuator Predicted Performance Summary

Based on the measured performance of the AC 17920-43 fluid amplifier cascade and power cylinder design parameters, the following performance predictions and comparisons with design goal objectives were made:

Performance Parameter	Contract Design Goal	Predicted Result
Actuator Stroke (in.)	± 0.200	± 0.200
Min. Output Force (lb)	2	2
Stiffness (operating) (lb/in.)	irreversible	106
Stiffness (no pressure)	---	irreversible
Scale Factor (in./psid)	0.10	0.0377
Signal for full stroke (psid)	± 2	± 5.35
Supply Flow (max.) ($in.^3$/sec)	1.0	2.03
Supply Pressure (psid)(nom.)	200	400
Dynamic Response	3 Hz, -3 dB bandwidth	1.2 Hz, -3 dB bandwidth

General Conclusions

1. Other than the low friction requirements of the power cylinder, servoactuator performance is determined by the operational characteristics of the driver fluid amplifier or cascade. Because relatively constant cascade gain over the operating fluid temperature range is required, it was concluded that operation in the turbulent flow regime is the most practical approach.

2. The 1.0 in.^{3}/sec maximum power flow requirement is impractical to achieve servoactuator compatibility (scale factor) with existing HYSAS controllers. This supply flow limitation also makes it highly doubtful that the power amplifier can be designed in a practical size and achieve the desired output resistance (P-Q curve slope).

3. Without resorting to an extensive fluid amplifier development effort, it was concluded that the servoactuator predicted performance could be upgraded to a 2 Hz, -3 dB bandwidth response and be compatible with existing HYSAS fluidic controllers if the supply flow limit of 1.0 in.^{3}/sec was raised to approximately 3.0 in.^{3}/sec at nominal test conditions.

4. There is little likelyhood of meeting the servoactuator performance goals as specified in the contract. Therefore, the technical effort was redirected to the alternate servoactuator design described in the next section.

ALTERNATE SERVOACTUATOR CONCEPT

As a result of the problems encountered with the initial servoactuator design, an alternate mechanization was defined. The components of the alternate design are the same as those used in the initial design with the addition of a conventional spool valve. The power fluidic amplifier and the spool valve are combined to obtain a pressure control servovalve that drives the spring-centered power cylinder. The advantages of the addition of the spool valve are: (1) it provides the

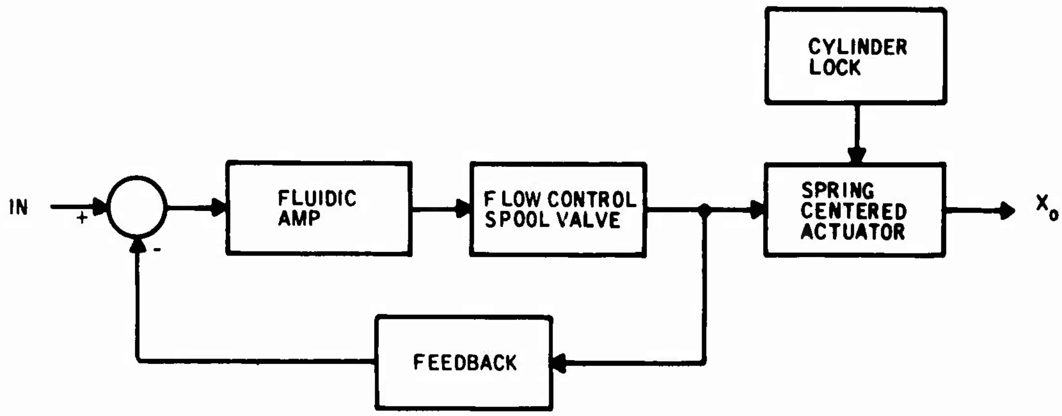
flow gain and low output impedance needed to provide acceptable actuator response without high supply flow leakage, and (2) because of the small spool motions, higher amplifier output impedance can be tolerated without producing significant lags in the servoactuator.

Figures 6 and 7 show two approaches to the alternate servoactuator design. The two configurations are the same with the exception of the type spool valve used. In the configuration shown in Figure 6, a flow control spool is used with pressure feedback around both the fluidic amplifiers and the spool valve. In the pressure control spool configuration shown in Figure 7, the primary feedback is internal to the spool valve. However, some feedback is also used around the amplifier cascade for final scale factor adjustment. The following subsections summarize the servoactuator design analysis performed for the two configurations.

General Design Considerations

The general design parameters for the servoactuator are:

Scale Factor	0.1 in./psid
Stroke	±0.2 in.
Estimated Friction	2 lb
Piston Area (A_p)	0.2 in.^{2}



BLOCK DIAGRAM

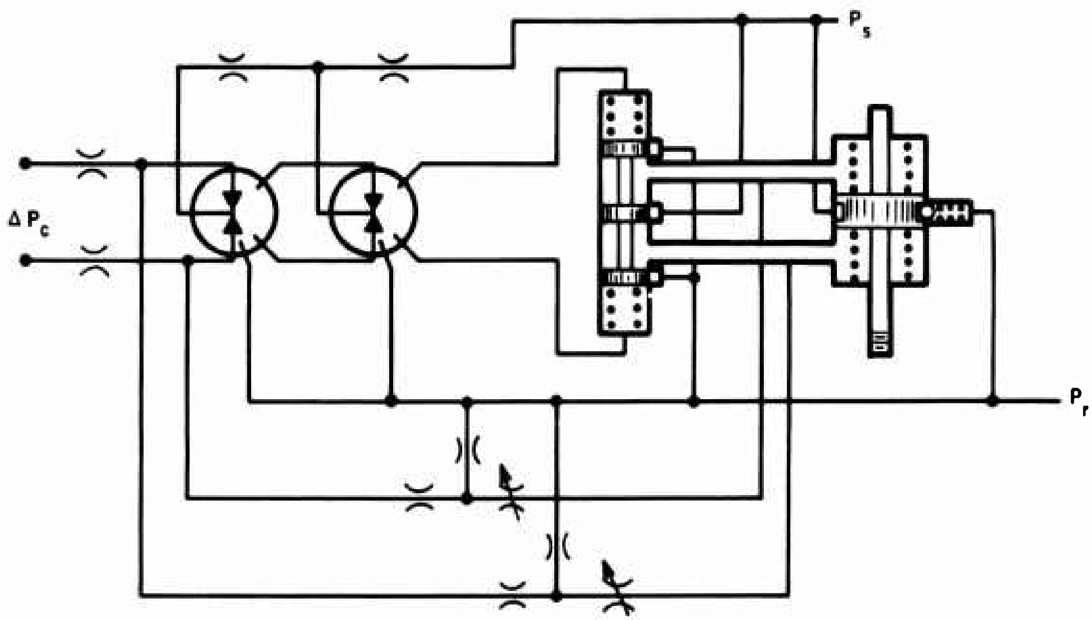


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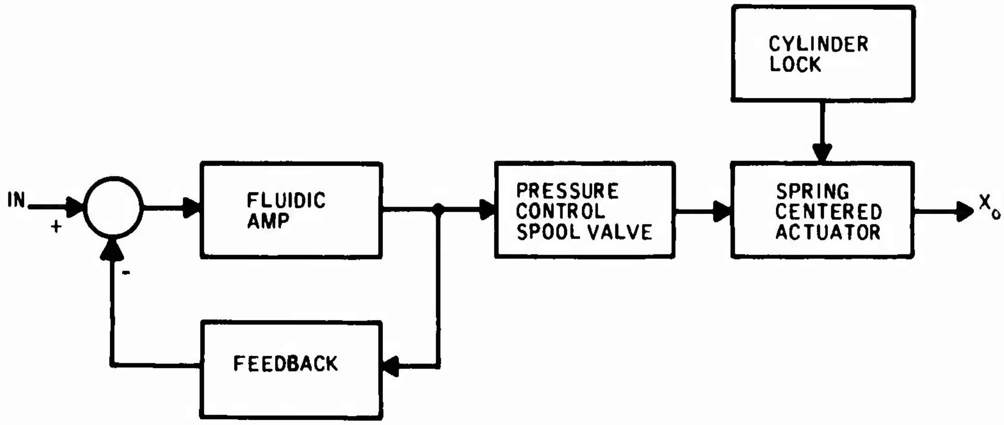
Figure 6. Servoactuator — Flow Control Spool Configuration



BLOCK DIAGRAM

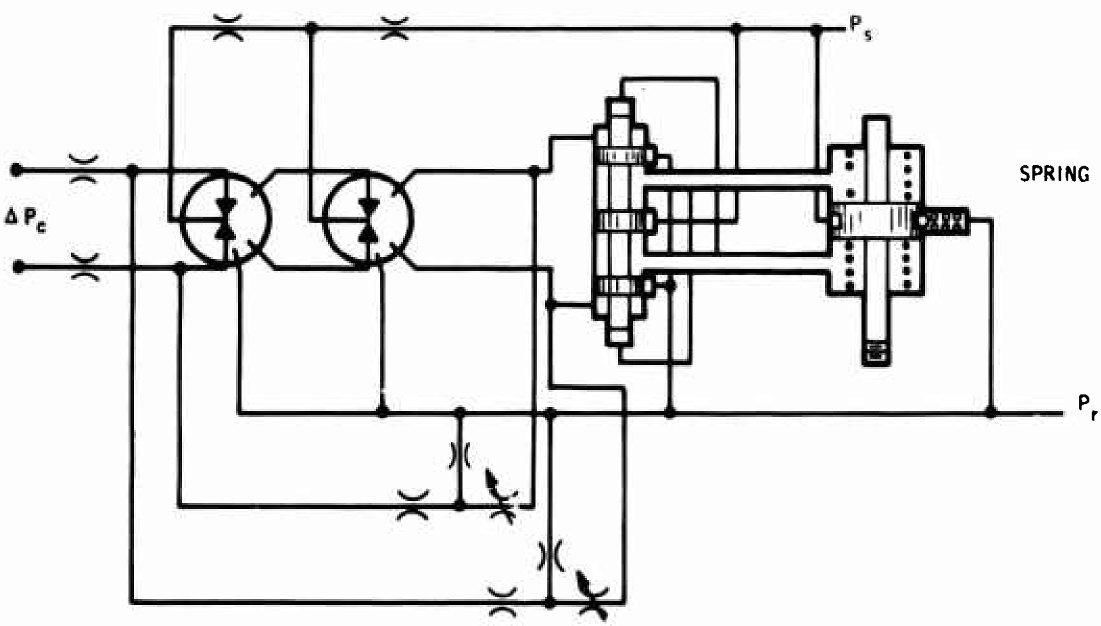


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Figure 7. Servoactuator — Pressure Control Spool Configuration

The following table lists: (1) the maximum spool output pressure required, ΔP_o (max.); (2) the maximum piston force due to spool pressure, F_p (max.); and (3) the actuator deadband due to friction for several spring rate values of the cylinder centering springs, K_s.

	Spring Rate (lb/in.)		
	100	150	200
ΔP_o(max.) -psid	100	150	200
F_p(max.) - lb	20	30	40
Deadband - %	10	6.7	5

As shown, increased centering spring rate results in lower deadband due to friction, but requires higher output pressures, which means higher servovalve pressure gain required. A nominal spring rate of 200 lb/in. was selected.

The minimum spool flow requirements are based on the cylinder velocity required for 50 percent amplitude at a frequency of 3 Hz.

$$X = (0.5)(0.2) \sin 18.8t$$

$$V = 1.88 \cos 18.8t \text{ or } 1.88 \text{ in./sec (max.)}$$

Using 2 in./sec velocity gives a minimum flow requirement of $2 \times 0.2 = 0.4$ in.^{3}/sec.

Flow Control Spool Configuration

The preliminary analysis of this configuration using a free spool indicated that the servoactuator would have very low damping. Therefore, a spring centered spool was selected for this design. The simplified block diagram shown in Figure 8 was used to determine the transfer function for the servoactuator.

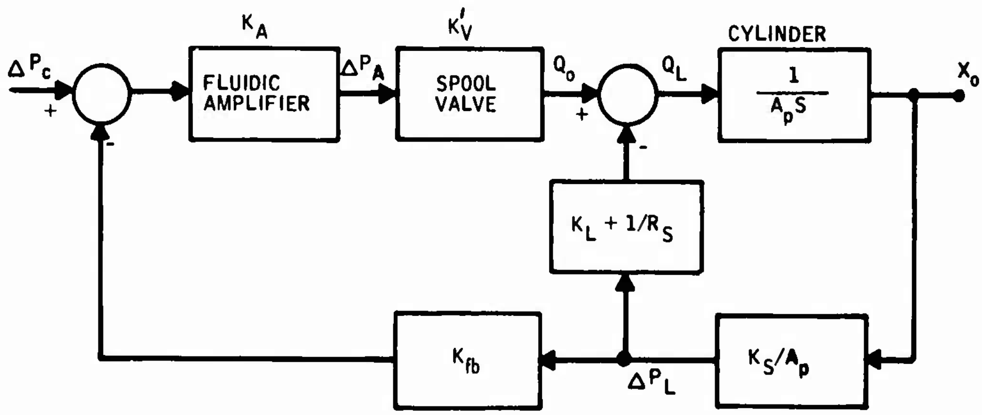


Figure 8. Block Diagram Used to Determine the Transfer Function for the Servoactuator Using the Flow Control Spool Configuration

where:

K_a	= Amplifier Pressure Gain (psi/psi)	To be determined (TBD)
K_v'	= Spool Valve Flow Gain (in.^{3}/sec/psi)	To be determined (TBD)
A_p	= Cylinder Area (in.^{2})	0.2
K_s	= Centering Spring Rate (lb/in.)	200
K_{fb}	= Pressure Feedback Gain (psi/psi)	To be determined (TBD)
K_l	= Cylinder Leakage (in.^{3}/sec/psi)	$\approx 5 \times 10^{-4}$
R_s	= Spool Valve Output Resistance (psi/in.^{3}/sec)	≈ 400

The transfer function is

$$\frac{X_o}{\Delta P_c} = \frac{K_a K_v{}^1 A_p}{A_p{}^2 S + K_s (K_1 + {}^1/R_s) + K_a K_v{}^1 K_{fb} K_s}$$

Using the parameter values from above, the scale factor and time constant are

$$S.F. = \frac{K_a K_v{}^1 A_p}{K_s (K_1 + {}^1/R_s) + K_a K_v{}^1 K_{fb} K_s} = \frac{0.2\ K_a K_v{}^1}{200 K_a K_v{}^1 K_{fb} + 0.6}$$

$$T = \frac{A_p{}^2}{K_s (K_1 + {}^1/R_s) + K_a K_v{}^1 K_{fb} K_s} = \frac{0.04}{200 K_a K_v{}^1 K_{fb} + 0.6}$$

Assuming a scale factor of 0.1 in./psi and a time constant of 0.05 second are desired, the gain values required are

$$K_a K_v' = 0.4 \text{ in.}^3/\text{sec/psi} \qquad K_{fb} = 0.0025 \text{ psi/psi}$$

It was concluded that the described actuator design provides adequate response with reasonable gain requirements. Values of K_v' of 0.01 in.^{3}/sec/psi are obtainable from standard spring-centered spools, which means a fluidic amplifier gain of 40 would be required.

Pressure Control Spool Configuration

The simplified block diagram shown in Figure 9 is used to determine the transfer function for the servoactuator using a pressure control spool. Secondary effects, such as the dynamics between the amplifier cascade and the spool and the cylinder mass, friction, and leakage, are neglected for this analysis.

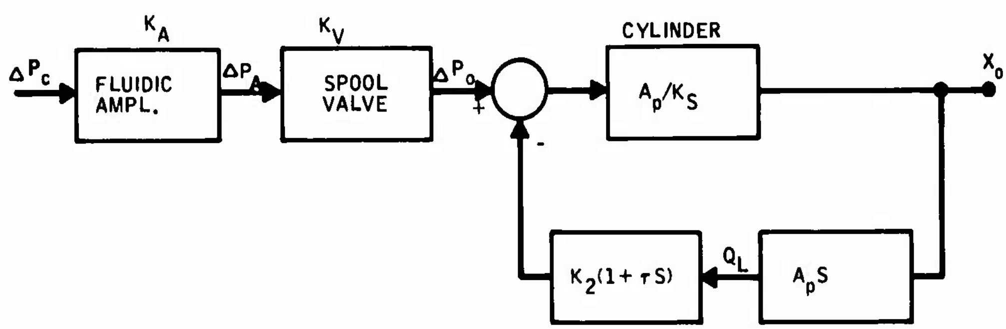


Figure 9. Block Diagram Used to Determine the Transfer Function for the Servoactuator Using a Pressure Control Spool

The servoactuator transfer function is:

$$\frac{X_o}{\Delta P_c} = \frac{K_a K_v A_p}{K_2 A_p^{\ 2} \tau S^2 + K_2 A_p^{\ 2} S + K_s}$$

Using the following parameter values

$A_p = 0.2$ in.^{2}
$K_s = 200$ lb/in.
$K_2 = 50$ psi/in.^{3}/sec ⎫ from Moog technical bulletin
$\tau = 0.016$ second ⎭ #103

The transfer function becomes:

$$\frac{X_o}{\Delta P_c} = \frac{6.25\, K_a K_v}{S^2 + 2\,(.4)\,79S + (79)^2}$$

For the required scale factor of 0.1 in./psid

$$K_a K_v = 100 \text{ psi/psi}$$

It was concluded that the described actuator design has more than adequate response and that the pressure gain requirements are within present capability. A typical pressure control spool has a pressure gain of about 2.5, which means a gain of 40 is required in the fluidic amplifier cascade.

Component Design

Fluidic Amplifiers - The fluidic amplifier cascade requirements are based on the requirements for the two configurations.

- Gain - Amplifier pressure gain should be at least 40. This should be obtainable with two amplifier stages.

- Output Range - For the pressure control spool, the output range is the spool maximum output pressure (200 psi) divided by the spool pressure gain (2.5) or a requirement of 80 psid. For the flow control spool, the required range is the spool flow range (0.4 in.^{3}/sec) divided by spool flow gain (0.01 in.^{3}/sec/psi) or 40 psid.

- Output Impedance - The output resistance of the fluidic amplifier combined with the capacitance of the spring-centered spool produced a first-order lag in the servoactuator. It is assumed that the time constant of this lag should be less than 0.01 second. Using a value of 10^{-5} in.^{5}/psi for the spool capacitance, the amplifier output impedance must be less than 1000 (lb-sec)/in.^{5}. Based on earlier tests with a 0.020-by 0.020-in.-size amplifier, which has an output resistance of approximately 300(lb-sec)/in.^{5}, a minimum amplifier size of 0.011 by 0.011 in. was calculated. Therefore, a 0.015-by 0.015-in. -size amplifier was selected.

- Power Flow - Using the given amplifier size and a supply pressure of 200 psi, a power flow of 0.32 in.^{3}/sec is calculated. Assuming two stages, the total flow should be less than 0.65 in.^{3}/sec; this leaves 0.35 in.^{3}/sec for the spool valve, which should be adequate.

Data was taken on both single-stage and two-stage amplifiers of the FG1004AA06 design. This amplifier has a 0.015-by 0.015-in. power nozzle and is the design used for an earlier fluidic SAS program. Figure 10 shows gain versus supply pressure data for this amplifier. The curve for two stages shows a gain variation between 88 and 132 over a supply pressure range of 50 to 300 psi. This is equivalent to a Reynolds number range of approximately 400 to 1400 or an increase by a factor of 3.5 to 1. Over the fluid temperature range of 40° to 180°F, the Reynolds number changes by a factor of 6 to 1 (at constant flow). This indicates that gain variation may be a problem. However, the feedback in the servovalve, particularly in the flow control spool

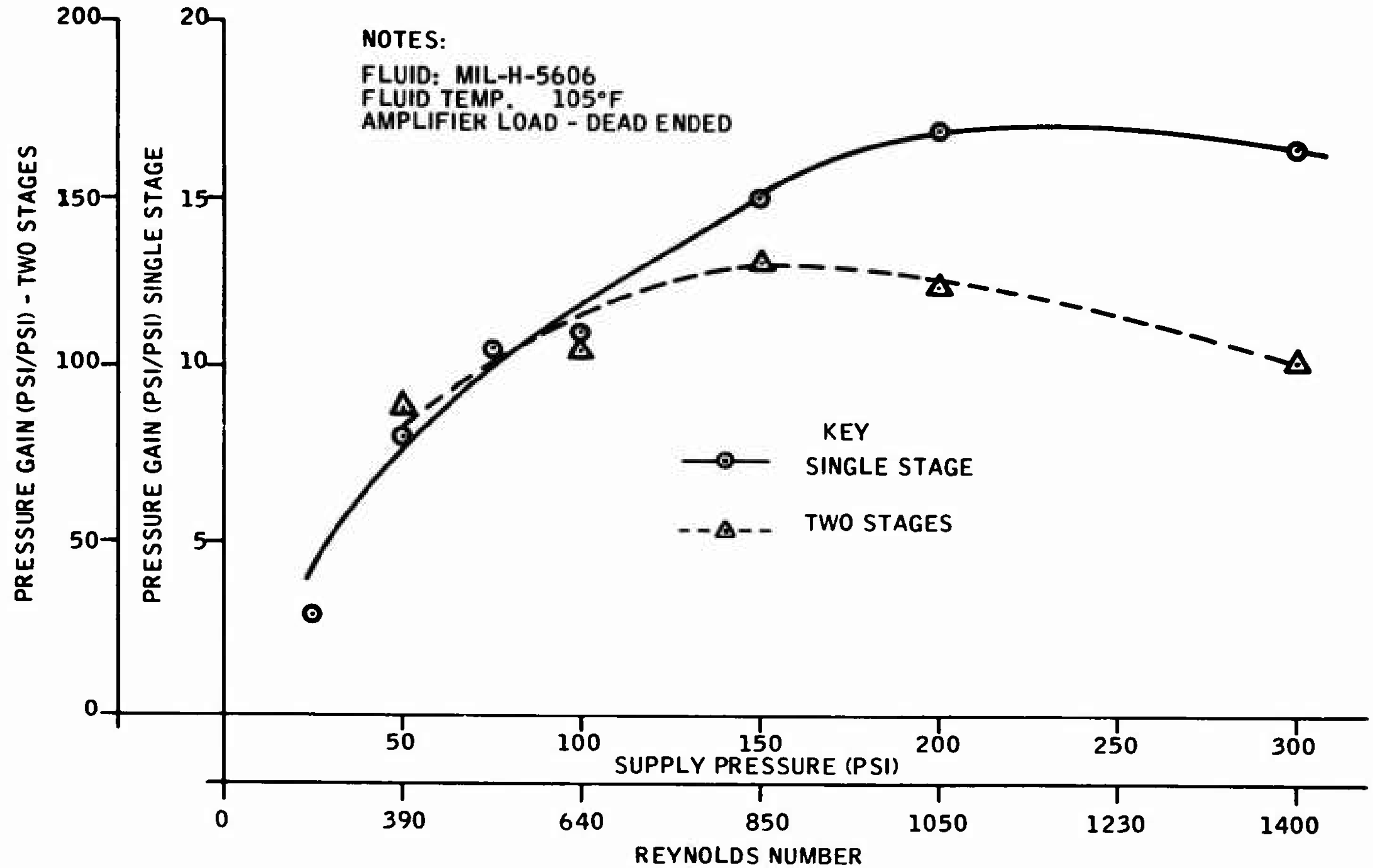


Figure 10. Pressure Gain of FG1004A106 Amplifier

configuration, will reduce the effect on the total servoactuator gain stability.

During testing of the servoactuator at low oil temperature, it was found that the gain of the FG1004AA06 amplifier decreased greatly. Part of this is due to the relatively long power nozzle of this amplifier, which creates a large viscous pressure loss at low oil temperature. A second amplifier design, the FG1004AA05, which is similar except for a shorter power nozzle, was substituted for the temperature testing performed later. Figure 11 is a picture of the two amplifier styles used during the program.

Spool Valves - Both units are standard designs available from Moog Inc. The flow control unit is a Series 21 valve with a spring-centered spool. The pressure control unit is a Series 15 valve. Table 1 summarizes the major performance characteristics of the two units. Figure 12 is a picture of the two spool valves.

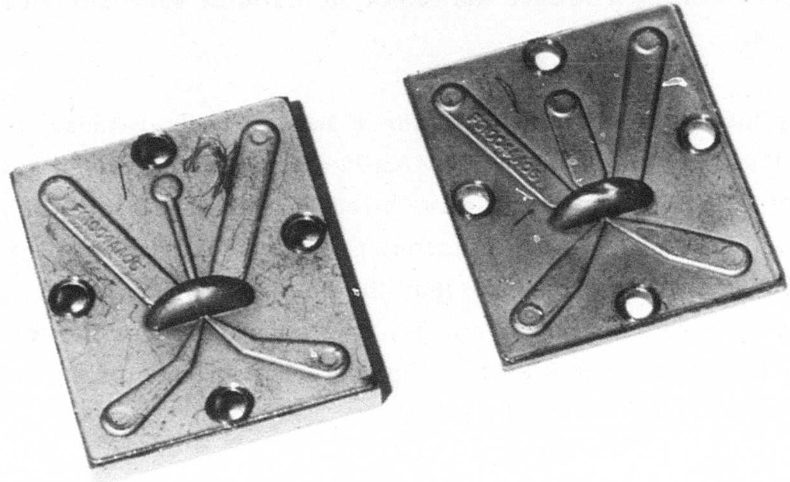
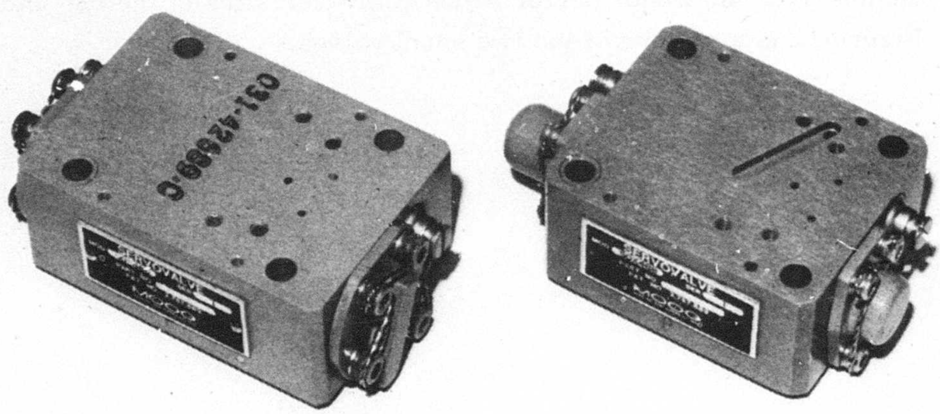


Figure 11. Fluidic Amplifier Elements



PRESSURE CONTROL SPOOL FLOW CONTROL SPOOL

Figure 12. Pressure Control and Flow Control Spool Valves

TABLE 1. MAJOR PERFORMANCE CHARACTERISTICS OF THE FLOW CONTROL AND PRESSURE CONTROL SPOOL VALVES

Parameter	Flow Control	Press Control
Supply Pressure (psi)	250	250
Rated Flow (in.^{3}/sec)	1.0	1.0
Flow Gain (in.^{3}/sec/psi)	0.008	--
Pressure Gain (psi/psi)	>50	3.3
Internal Leakage (in.^{3}/sec at 250 psi)	<0.25	<0.25
Output Impedance (psi/in.^{3}/sec)	<500	<50
Null Stability (%)	<10	<10

Spring-Centered Cylinder - Objectives of the power cylinder configuration concept are simplicity, low cost, and low seal friction. Figure 13 is a sketch of the power cylinder concept. The power cylinder consists of a cylinder body, piston/rod assembly, ball/spring detent (lock), end caps, and piston seals. Teflon rings backed with O-rings are used to provide low friction seals. The ball/spring detent provides the cylinder lock function.

To unlock the cylinder, supply pressure is admitted to the annular chamber on the piston (chamber between the piston seal lands). The supply pressure also acts on the underside of the ball detent, and the pressure difference across the ball causes it to be pushed up into the cylinder body, unlocking the cylinder. The cavity containing the detent spring is vented to return, to ensure that a differential pressure exists across the ball at all times when the cylinder is pressurized. The

annular chamber on the piston, detent spring cavity, and detent ball is sized to ensure that adequate differential pressure is provided across the detent ball to produce operation. The locking spool has an O-ring seal to minimize leakage through the locking device. The detent spring force is selected to be only large enough to ensure detent operation when the supply pressure is turned off.

From Figure 13 it is seen that the cylinder body has a straight-through bore with end caps. The centering springs are dropped into the cylinder and held in place by the end caps. Standard catalog springs are used, and the spring preload is controlled by selecting the appropriate cylinder, piston, and spring lengths. Because the piston and cylinder are sized with a small diametral clearance, the end caps do not have pilot plug ends, but are allowed to float.

The end cap screws are tightened selectively to ensure that minimum cylinder friction is obtained. This approach is selected to avoid the costly, close-tolerance parts manufacture required for piloted end cap installation. To facilitate field repair and rebuild, the end caps could be drilled and pinned after installation. This would provide end cap location for low cylinder friction without the selective screw tightening process initially used.

It is also noted that the detent ball and spring would be installed through an external opening in the side of the cylinder body. The detent would be installed after the cylinder had been assembled and mechanically checked for low friction.

Figure 14 shows the assembled cylinder with a position potentiometer attached to the output shaft for recording of cylinder position.

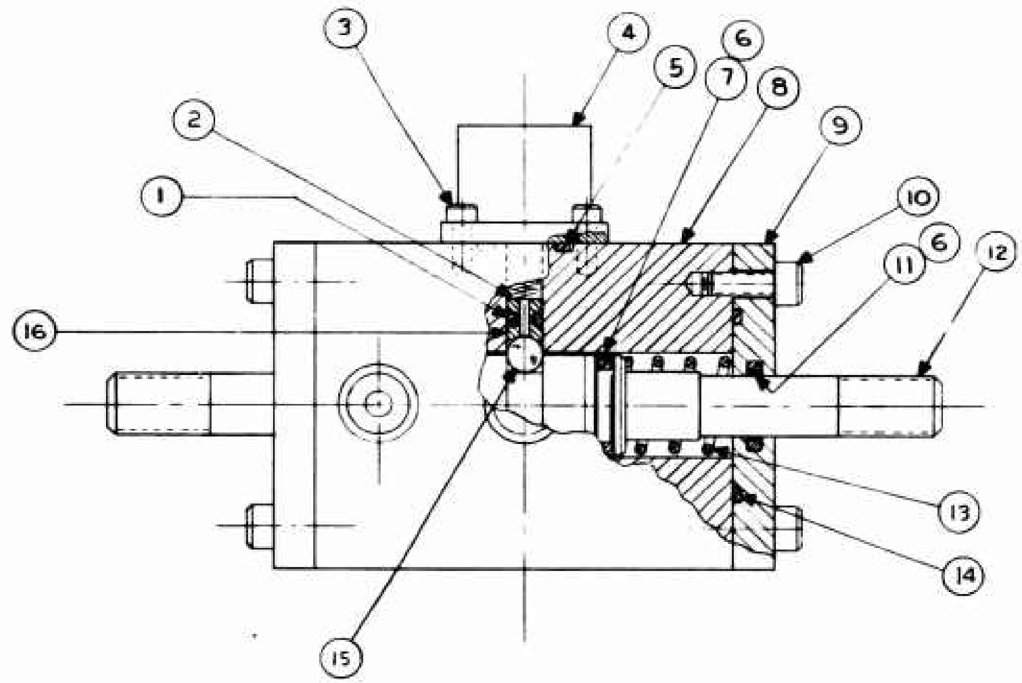


Figure 13. Spring-Centered Cylinder Concept

It should be noted that the springs shown on the two ends of the cylinder piston, outside of the cylinder body, are in addition to the internal centering springs. Standard springs, with a spring rate of 100 lb/in. each were not available, and, therefore, two springs, one internal and one external, were used on each end of the piston to obtain a total centering spring rate of 200 lb/in. Figure 15 shows the various parts of the cylinder assembly.

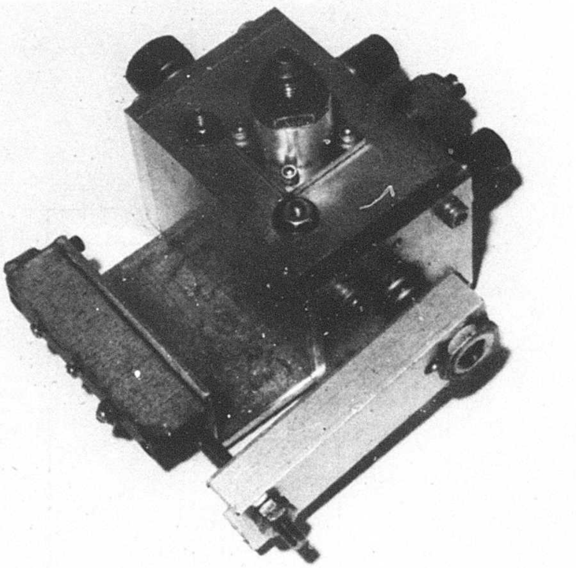


Figure 14. Spring-Centered Cylinder Assembly

Feedback Network - The feedback network consists of two small valves and four orifice resistors. They are incorporated in a small block similar to that used for feedback in some of the SAS circuits. The schematic diagrams of Figures 6 and 7 show the bridge configuration used for feedback signals. Each feedback path consists of

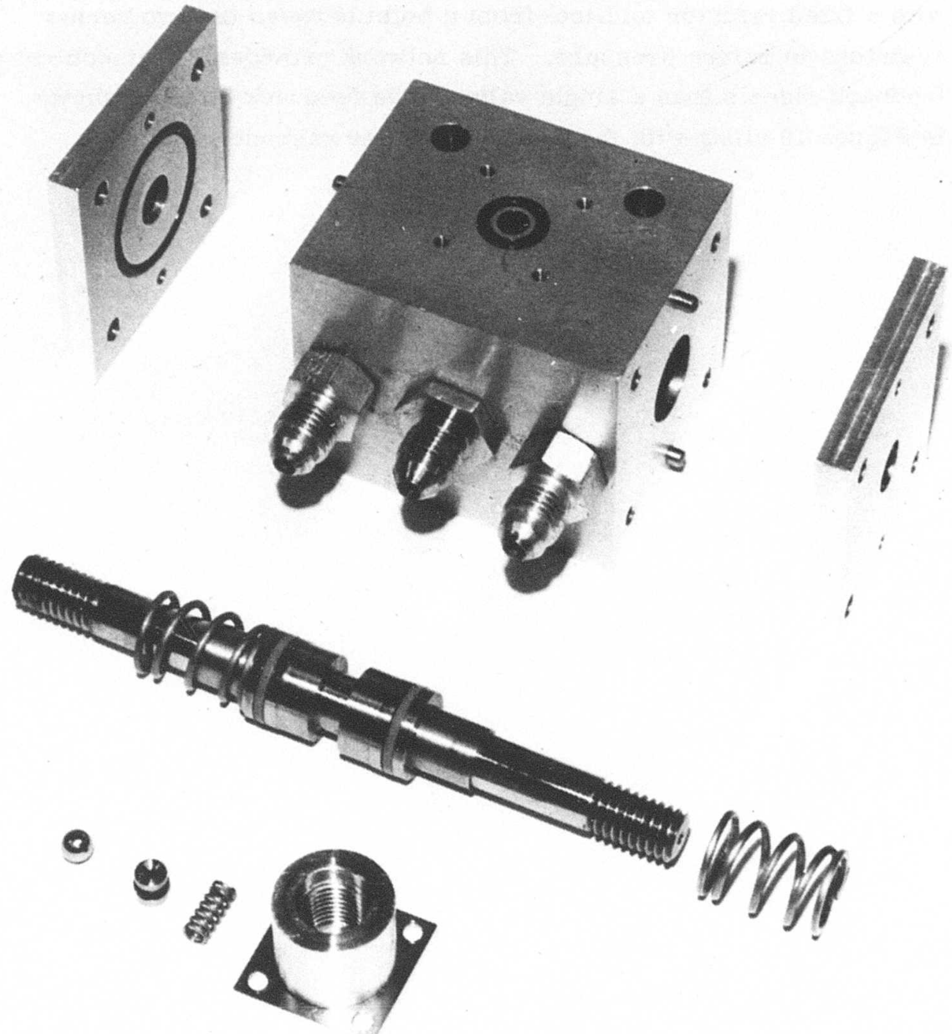


Figure 15. Spring-Centered Cylinder Components

a fixed resistor (orifice) in series with a variable resistor (valve) with a fixed resistor (orifice) from a point between the two series resistors to return pressure. This network provides better control of feedback signals than a single valve. The feedback block is shown in Figure 16 along with the fluidic amplifier cascade.

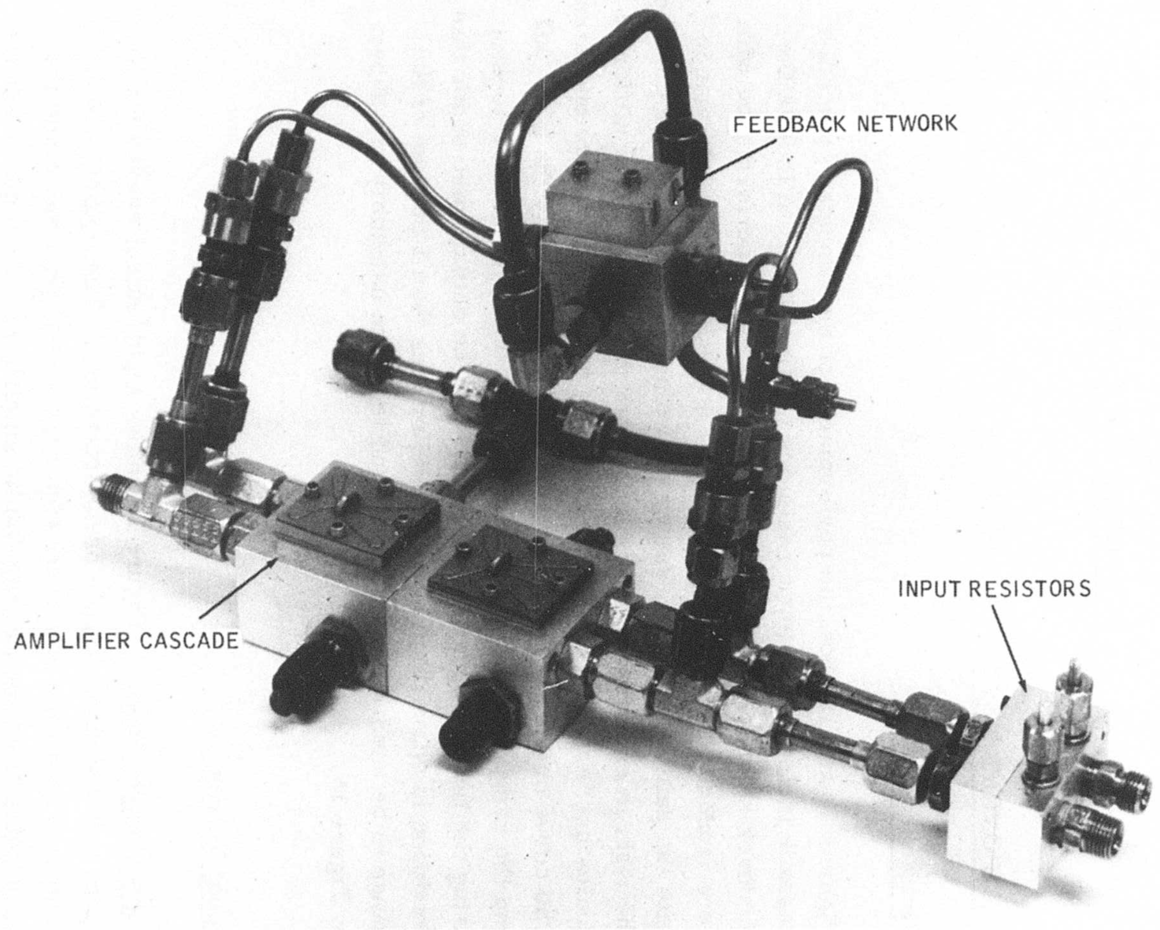


Figure 16. Fluidic Amplifier and Feedback Assembly

SECTION III
PERFORMANCE TESTS

COMPONENT TESTS

Fluidic Amplifiers

Figures 17 and 18 show the performance of the two-stage amplifier cascade including the input resistors (orifices) used to isolate the feedback signal from the input. Figure 17 shows performance with zero feedback (maximum gain), and Figure 18 shows performance with the gain reduced by a factor of two using feedback. The curve on the right in each figure shows the center part of the curve (different scales used) which would be used during normal servovalve operation. The curve on the right in Figure 18 has approximately the gain (35) and the range (±60 psid) required to drive the actuator stop-to-stop using the pressure control spool valve. The output noise under this condition is approximately ±0.5 psid or less than 1 percent of the range to be used. The amplifier and feedback assembly are shown in Figure 16.

Spool Valves

Flow Control Spool -- Figure 19 shows input/output curves for the spool valve at three supply pressure values. The output noise is associated with the relatively high gain of the spool valve and the input noise from the driving fluidic amplifiers. A supply pressure of 400 psi was selected for subsequent servoactuator testing. At this condition, the spool valve has a pressure gain of approximately 113.

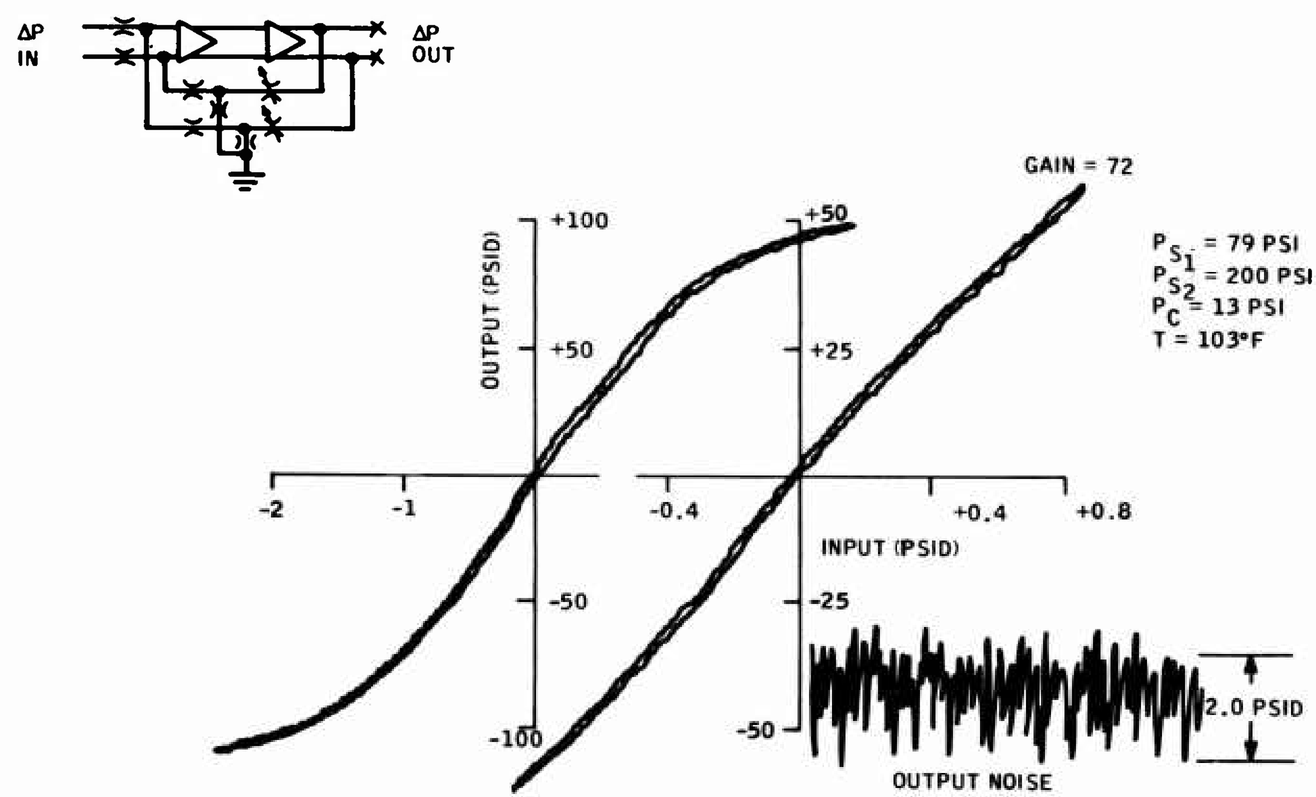


Figure 17. Amplifier Cascade Performance with Zero Feedback

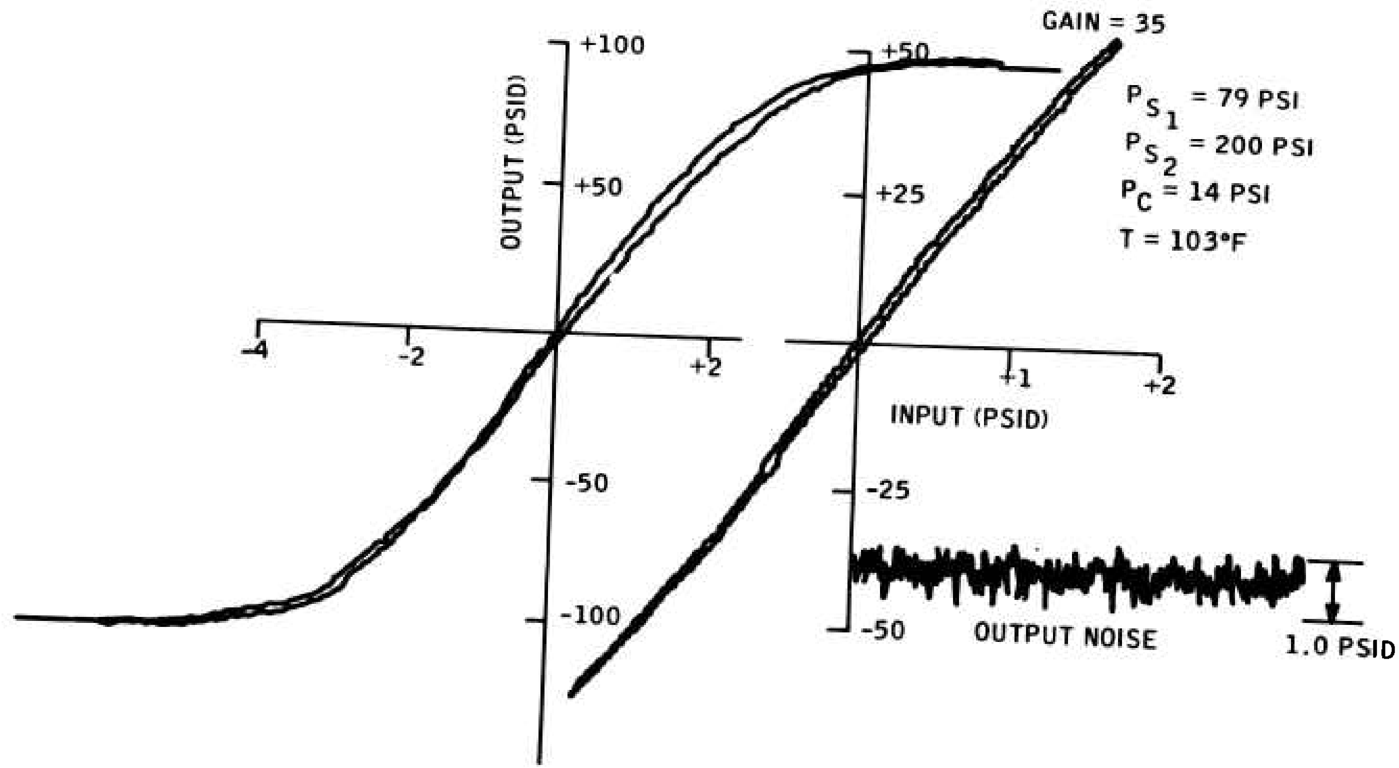


Figure 18. Amplifier Cascade Performance with Feedback

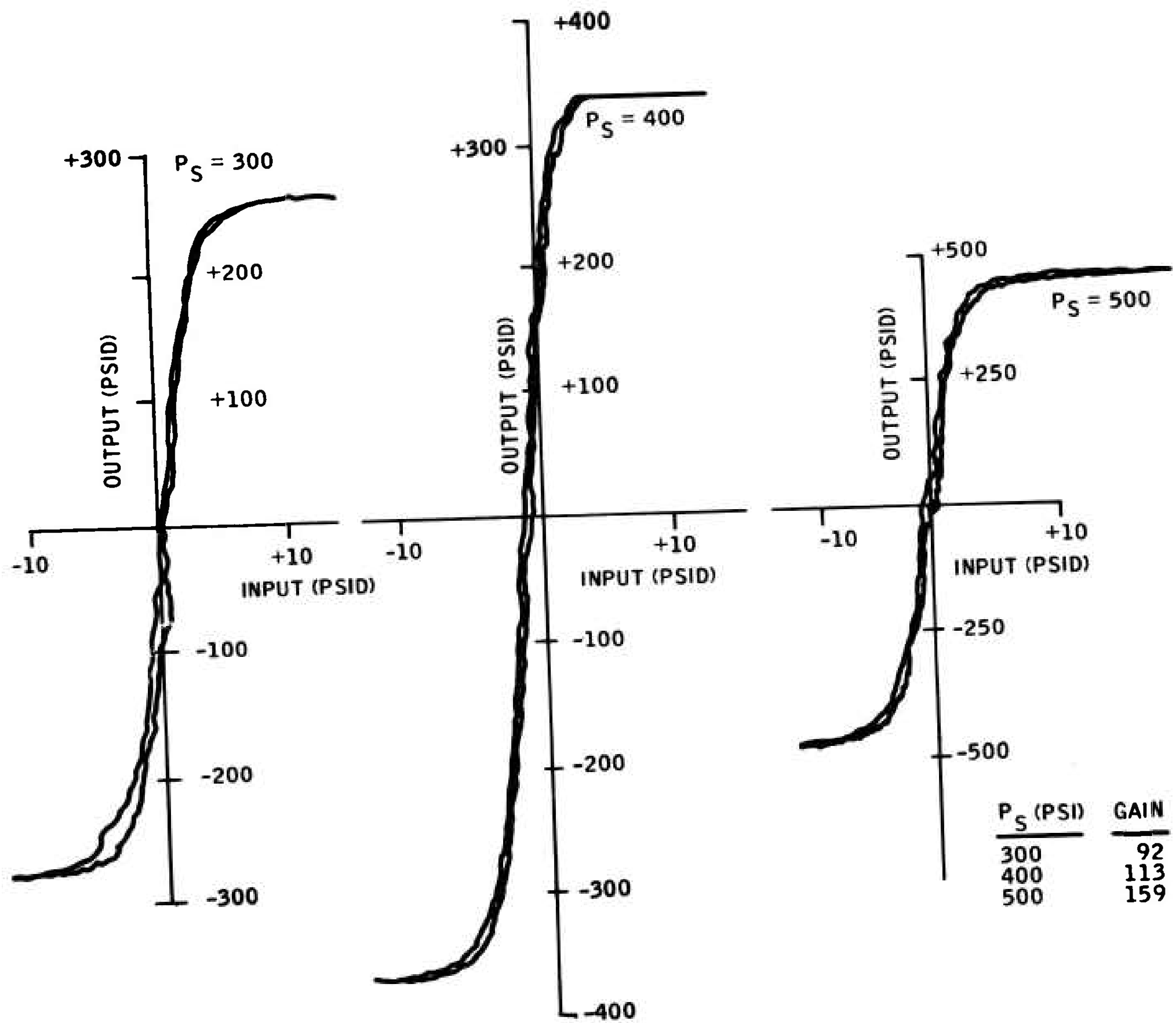


Figure 19. Flow Control Spool Valve Performance

Pressure Control Spool -- Figure 20 shows input-output curves for the pressure control spool valve at four supply pressures. The pressure gain of the valve is essentially constant at 3.2 psi/psi over the range of supply pressures tested. An output range of at least ±200 psi is required to move the cylinder to its full travel; therefore, a supply pressure of 400 psi was selected for this valve.

Spring-Centered Cylinder

Figure 21 shows the input-output curve for the spring-centered cylinder. Friction in the cylinder, which shows up as actuator deadband, was a problem. Two actions were taken to reduce the friction to the level shown in Figure 21: (1) the major diameter of the threads of the piston rod ends were cut down to avoid damaging (gouging) the rod end seals during installation, and (2) slightly smaller wire springs were installed inside the cylinder to prevent rubbing on the piston rod and cylinder walls. The total centering spring rate of 180 lb/in. and the cylinder area of 0.2 in.^{2} result in the cylinder scale factor of 0.001 in./psid. The curve on the right in Figure 21 shows the full output range of approximately ±0.20 inch, and the curve on the left shows approximately ±50 percent of the output range on an expanded scale. The deadband is ±5.6 percent of full scale-output, which is equivalent to a friction level of ±2.0 pounds.

The cylinder locking device was tested and found to operate satisfactorily. A supply pressure of approximately 70 psi is required to release the lock. In actual operation, the spool valve supply pressure of 400 psi is used to unlock the cylinder.

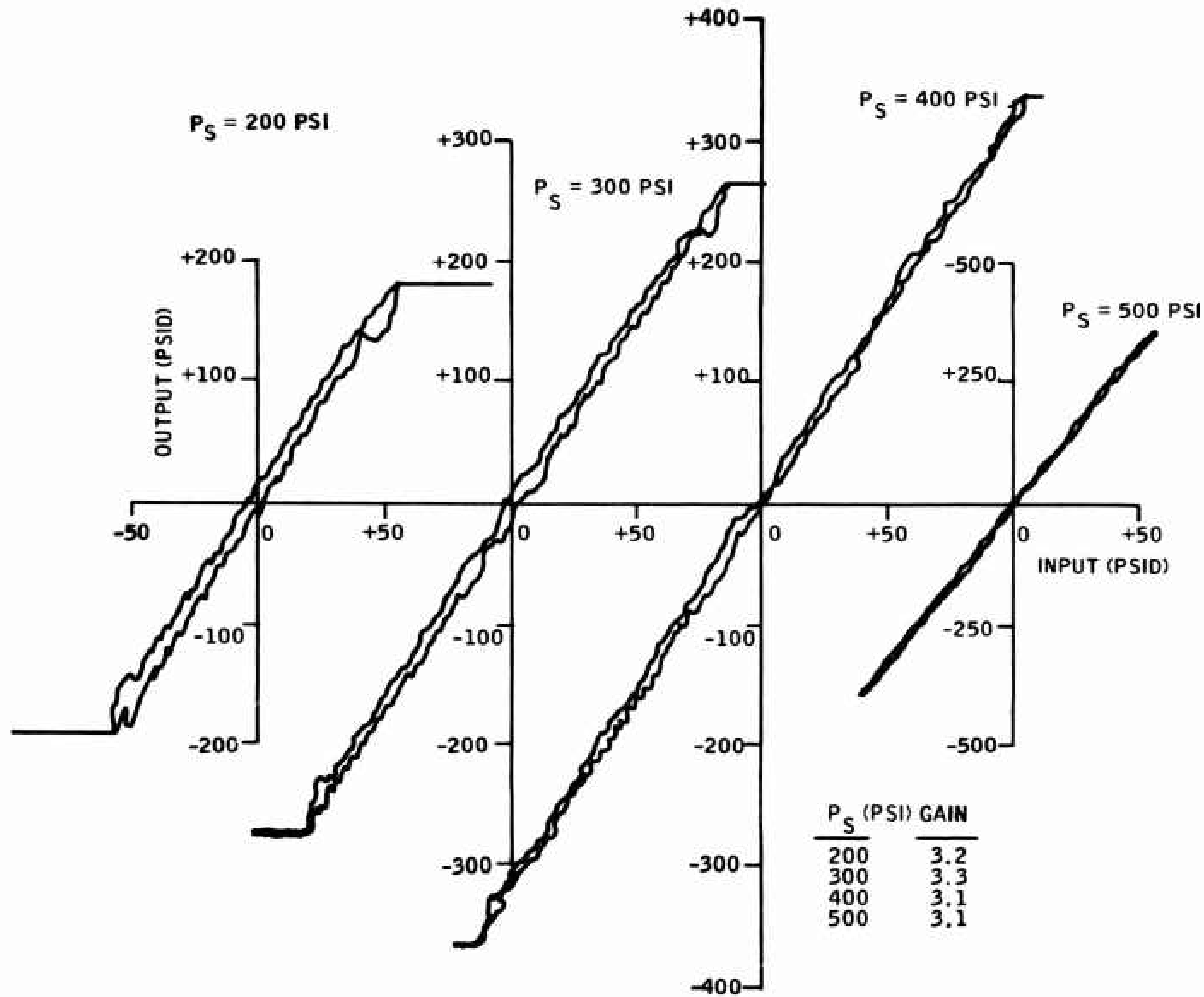


Figure 20. Pressure Control Spool Valve Performance

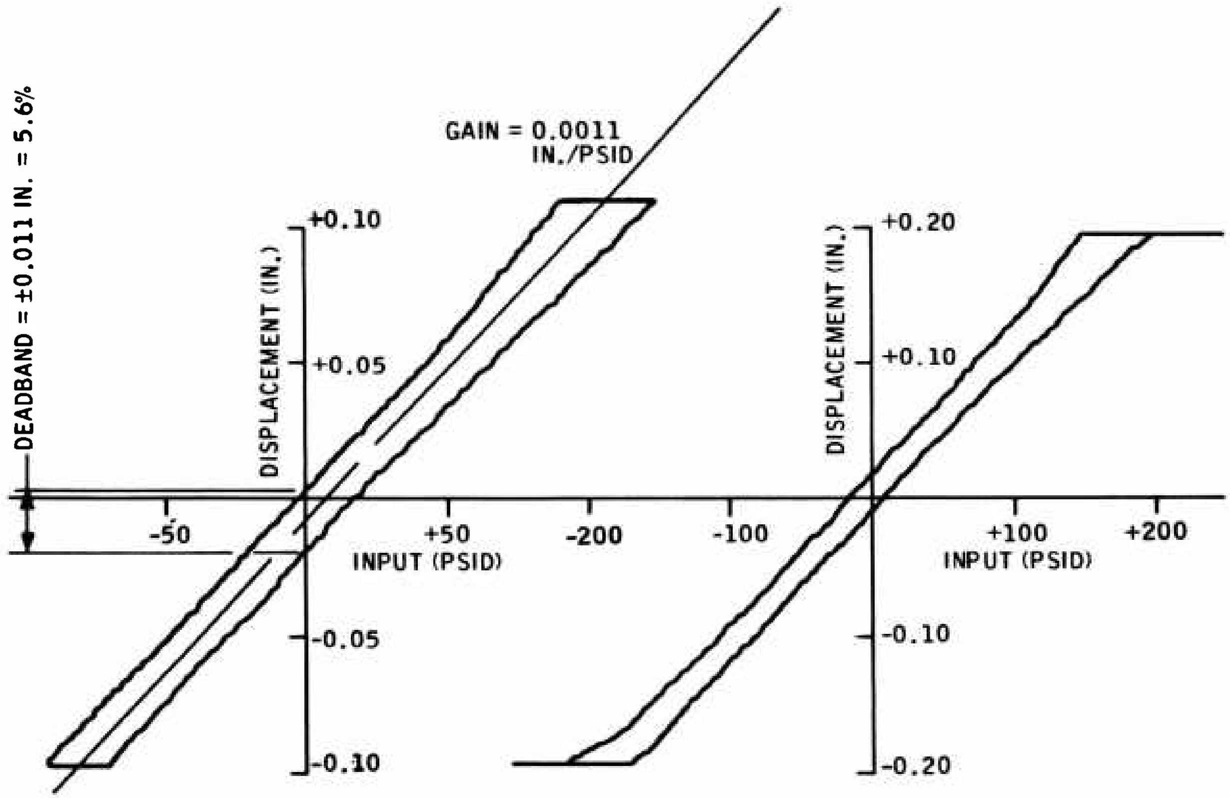


Figure 21. Spring-Centered Cylinder Performance

NOMINAL PERFORMANCE TESTS

Flow Control Spool Configuration

Figure 22 shows the gain of the servovalve (amplifiers and spool) and the gain of the servoactuator (amplifiers, spool, and cylinder) with the feedback adjusted to give a scale factor of 0.115 in./psid. In this case, the feedback has reduced the open loop gain by a factor of approximately 35. The actuator hysteresis has been reduced substantially for this circuit configuration, due to the noise which provides a dither signal to the cylinder.

Figure 23 shows variation in gain for change in supply pressure. A 20-percent change in supply pressure results in a 30-percent change in gain with an increase in supply pressure producing a gain decrease. This pressure sensitivity is due to feedback network gain change, which also is the cause of the fluid temperature sensitivity described later in this report.

Figures 24, 25, 26, and 27 show response curves for various parts of the circuit. Figure 24 shows the response of the amplifier cascade with the flow control valve spool as a load. The flow control spool has a spring-centered spool that is a capacitance load on the amplifier. This curve has, in addition to the time delay associated with the two amplifiers, a small first-order lag due to this capacitance. Figure 25 shows the response of the cylinder alone, and Figure 26 shows the response for the servovalve including the amplifier and the spool valve with feedback. The total servoactuator (which is the combination of the curves in Figures 25 and 26) is shown in Figure 27. The

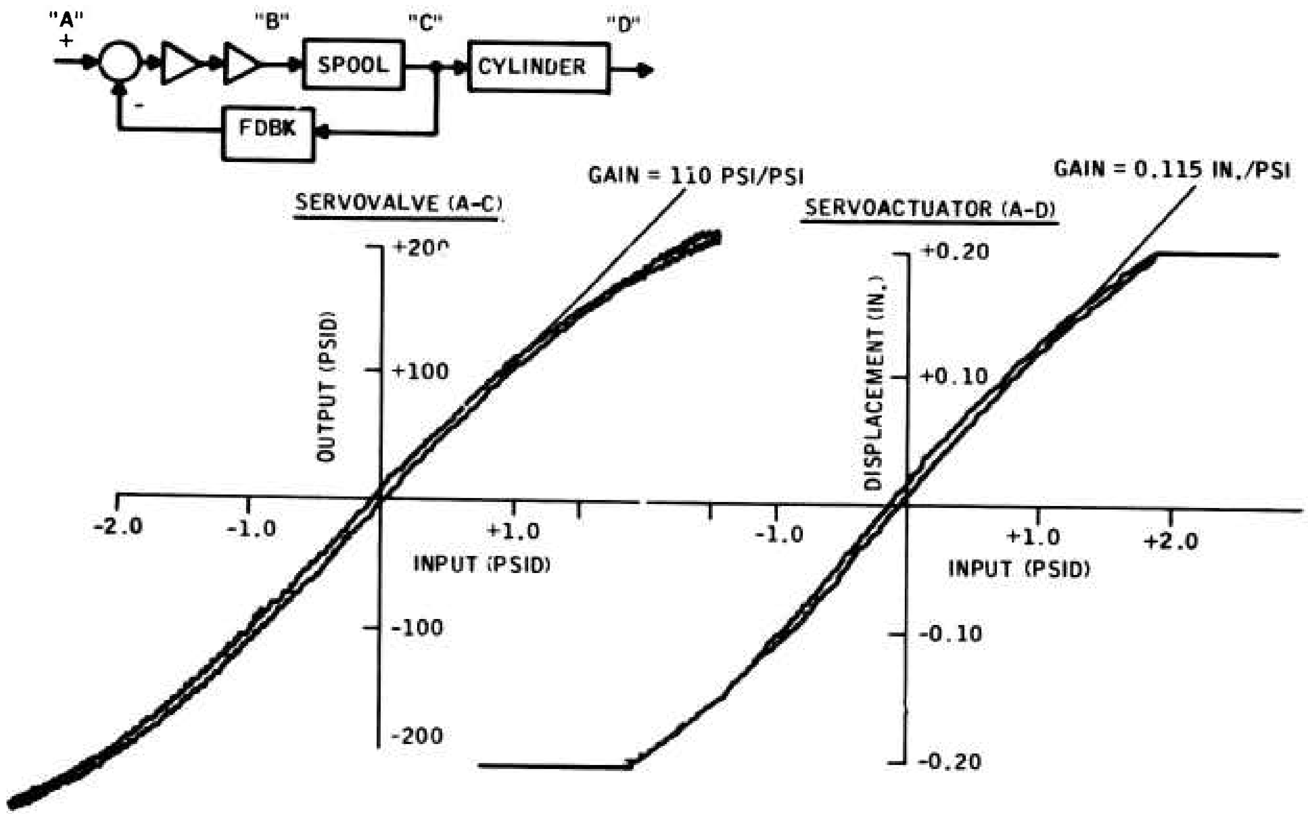


Figure 22. Servo Performance with Feedback (Flow Control Spool)

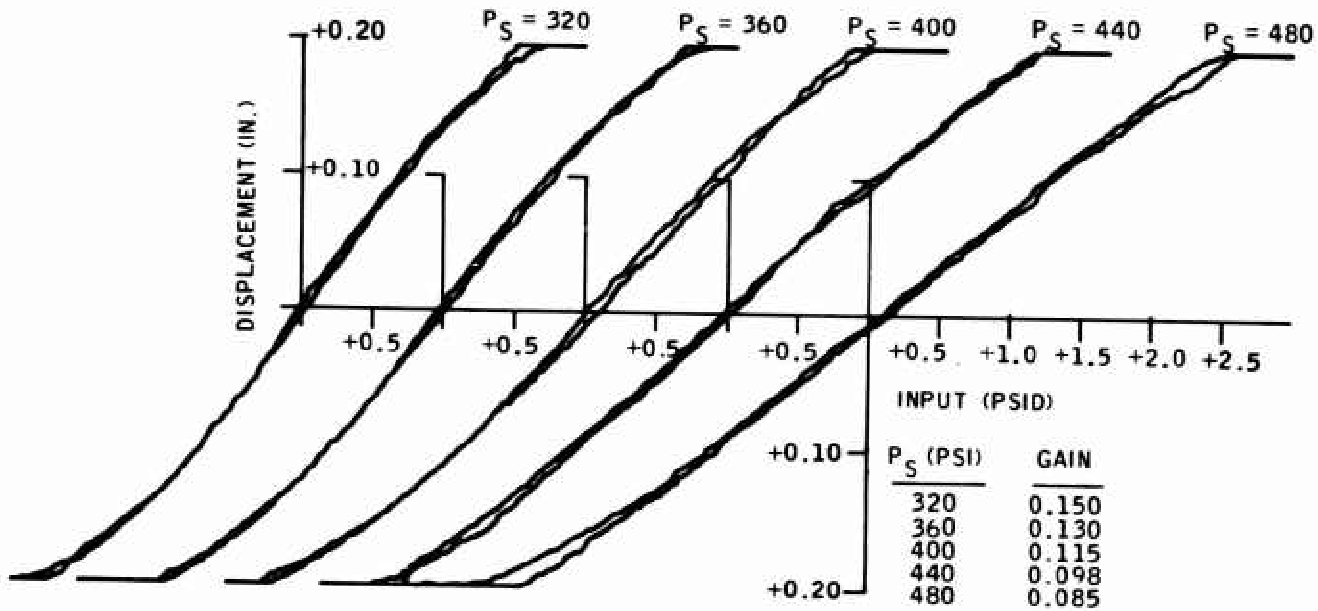


Figure 23. Servoactuator Performance Versus Supply Pressure

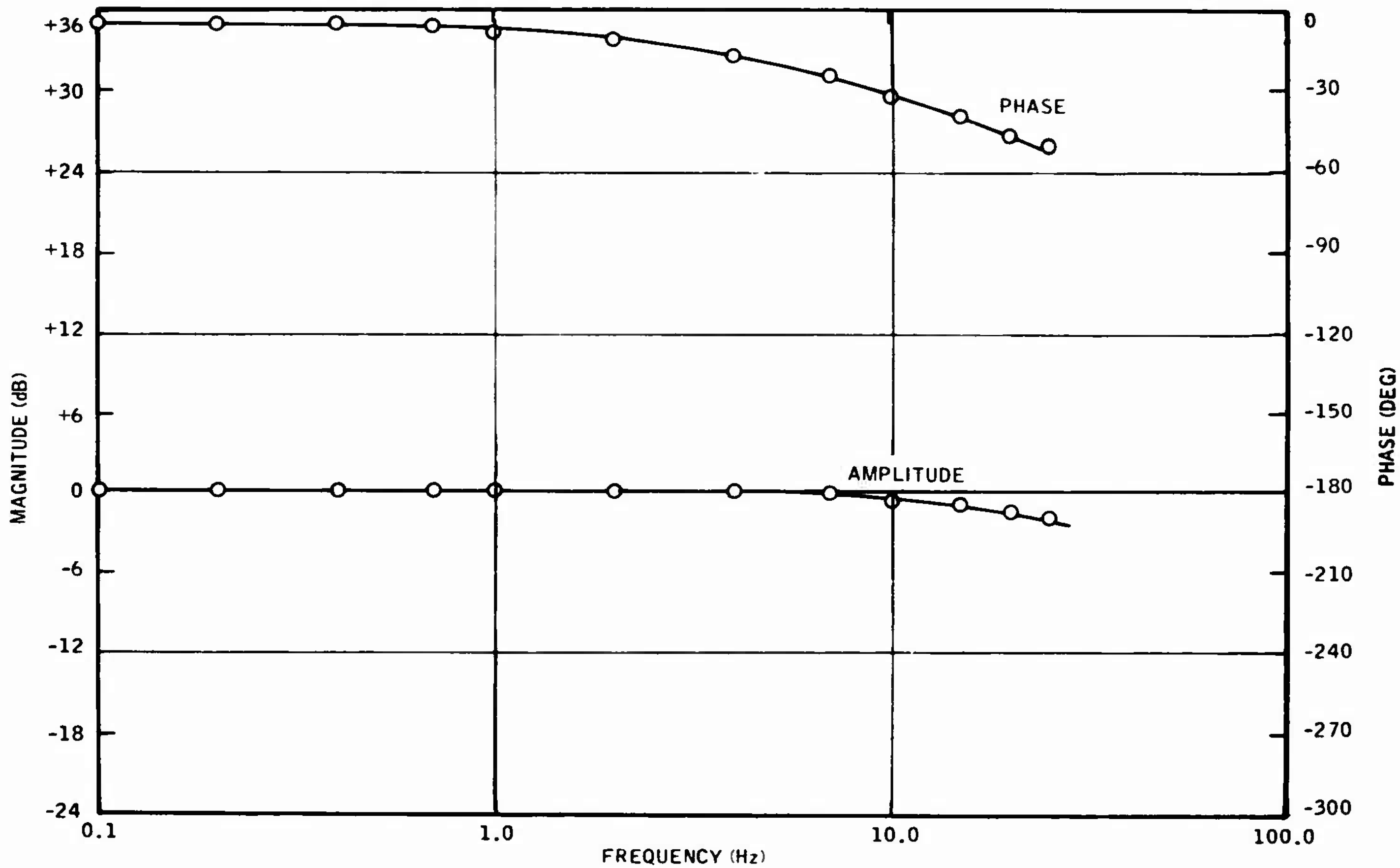


Figure 24. Fluidic Amplifier Response with No Feedback and Flow Control Spool as Load

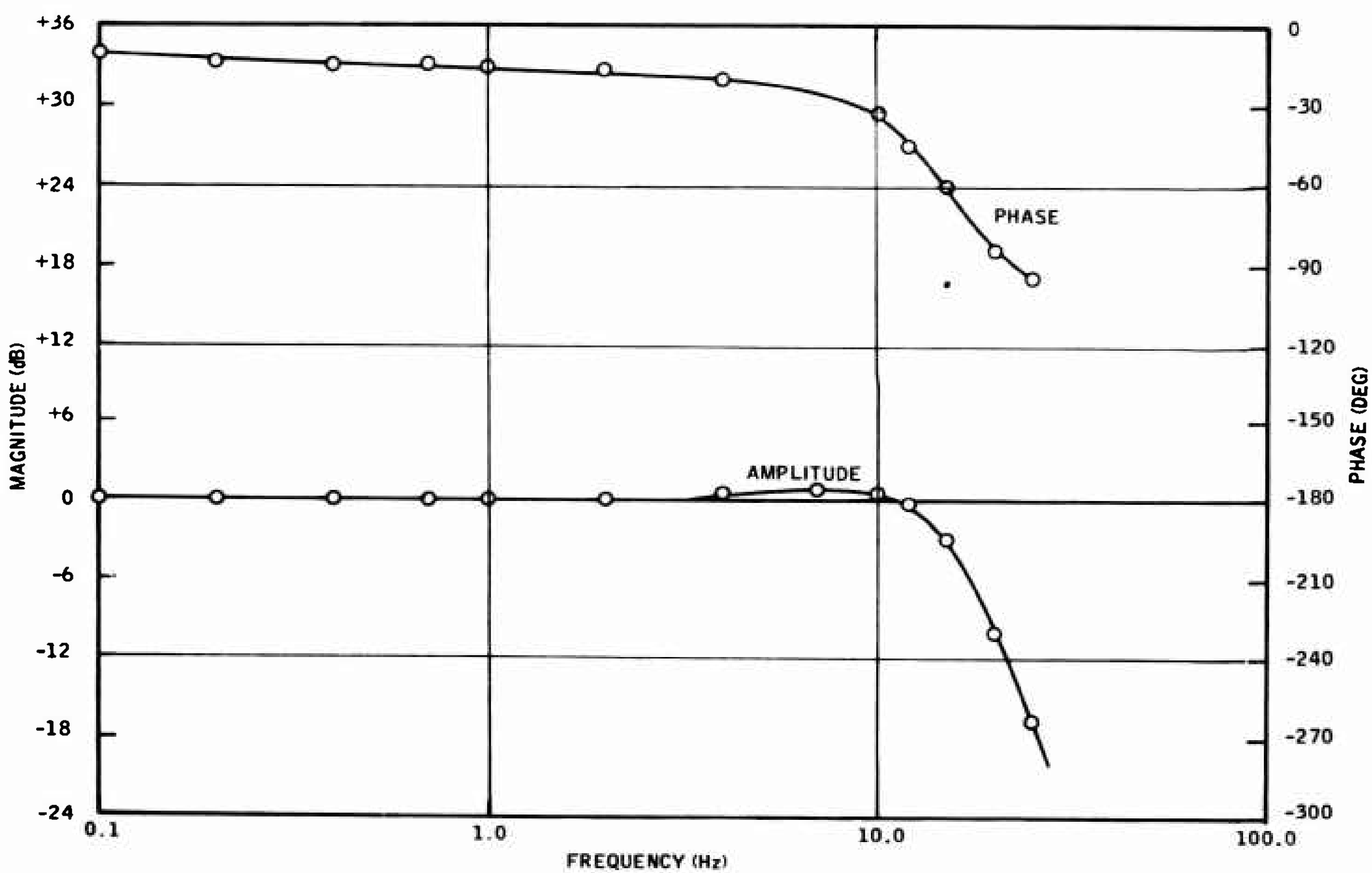


Figure 25. Spring-Centered Cylinder Response

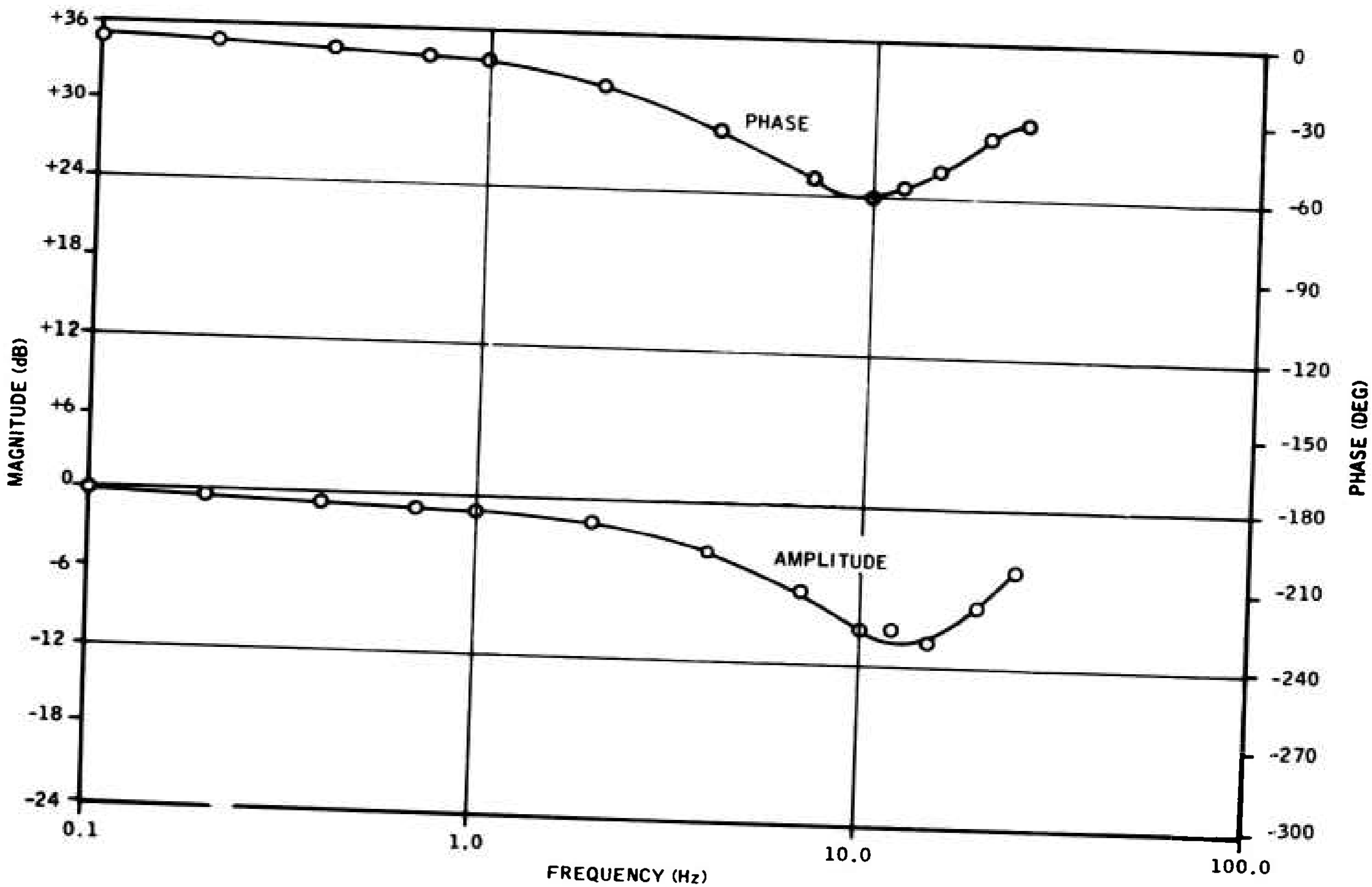


Figure 26. Servovalve Response (Flow Control Spool)

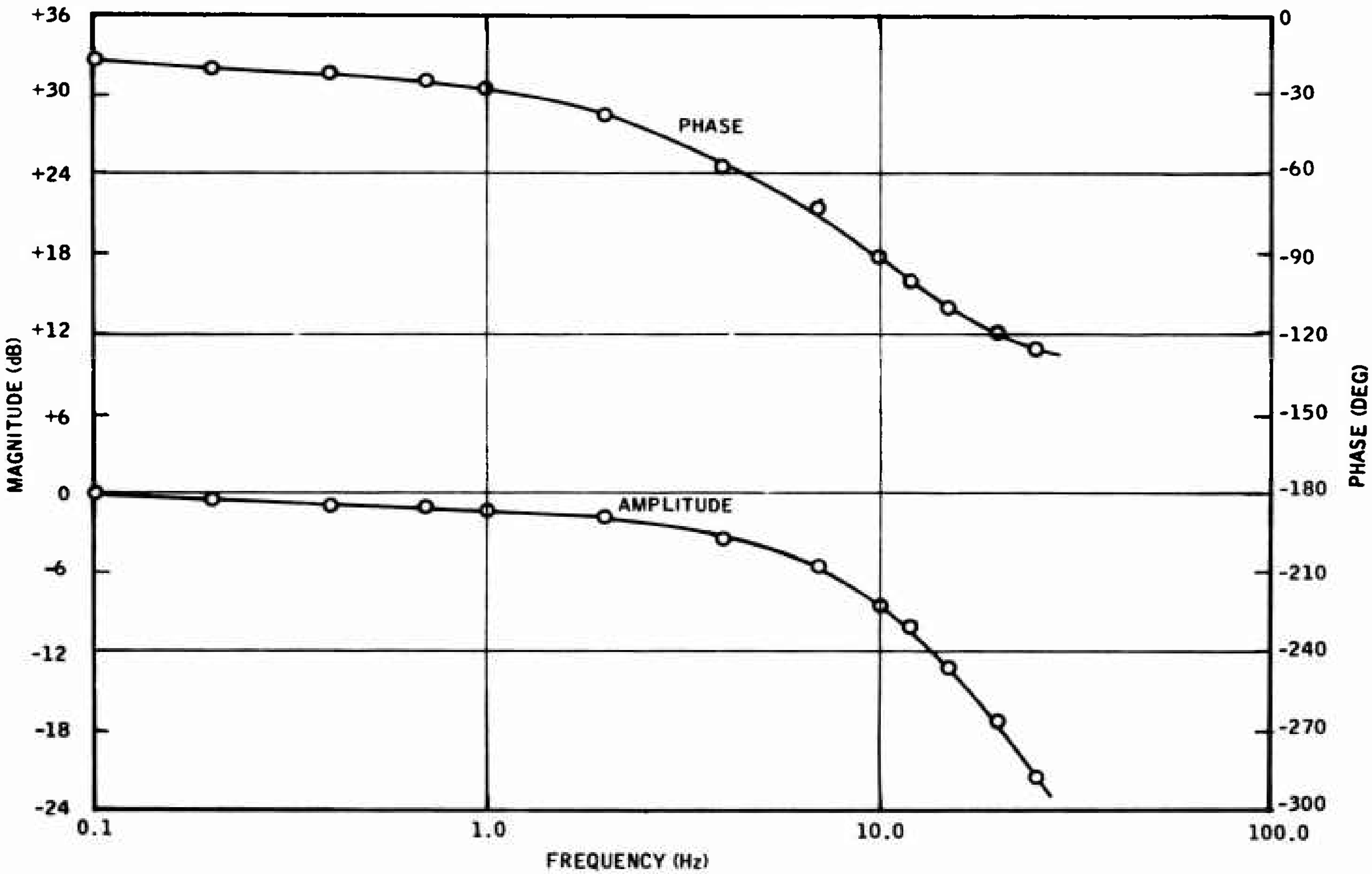


Figure 27. Servoactuator Response (Flow Control Spool)

servoactuator shows 3 dB attenuation at approximately 3. 5 Hz, which slightly exceeds the objective of 3 Hz.

Figure 28 shows the breadboard servoactuator assembly with the electric-to-fluidic input signal transducer. Tables 2 and 3 summarize the performance of the servovalve and servoactuator using the flow control spool at nominal supply conditions. The performance goals of the specification in Appendix A are given for comparison. In general, the servoactuator performance meets or exceeds the goals. The two marginal areas are linearity and load resistance. Linearity includes the effects of hysteresis and curvature of the gain curve. The principal reason for the relatively high nonlinearity is the curvature of the gain curve near full stroke. Linearity is considerably better over the center +50 percent of stroke. The actual load resistance value is slightly higher than the goal because the actual centering spring rate is approximately 10 percent lower than initially anticipated.

Pressure Control Spool Configuration

Initially, this configuration had the negative feedback coming from the output of the spool valve. However, problems of instability were encountered with this configuration; therefore, the feedback was changed to the configuration shown in Figure 7. It appeared that the friction in the cylinder combined with the cylinder capacitance resulted in pressure fluctuations causing a self-excited oscillation under certain conditions. Some of this effect occurs even with the feedback relocated due to the internal feedback within the spool valve from output to input. Figure 29 shows gain curves for the complete servoactuator and for the servovalve (sum point input to output of spool

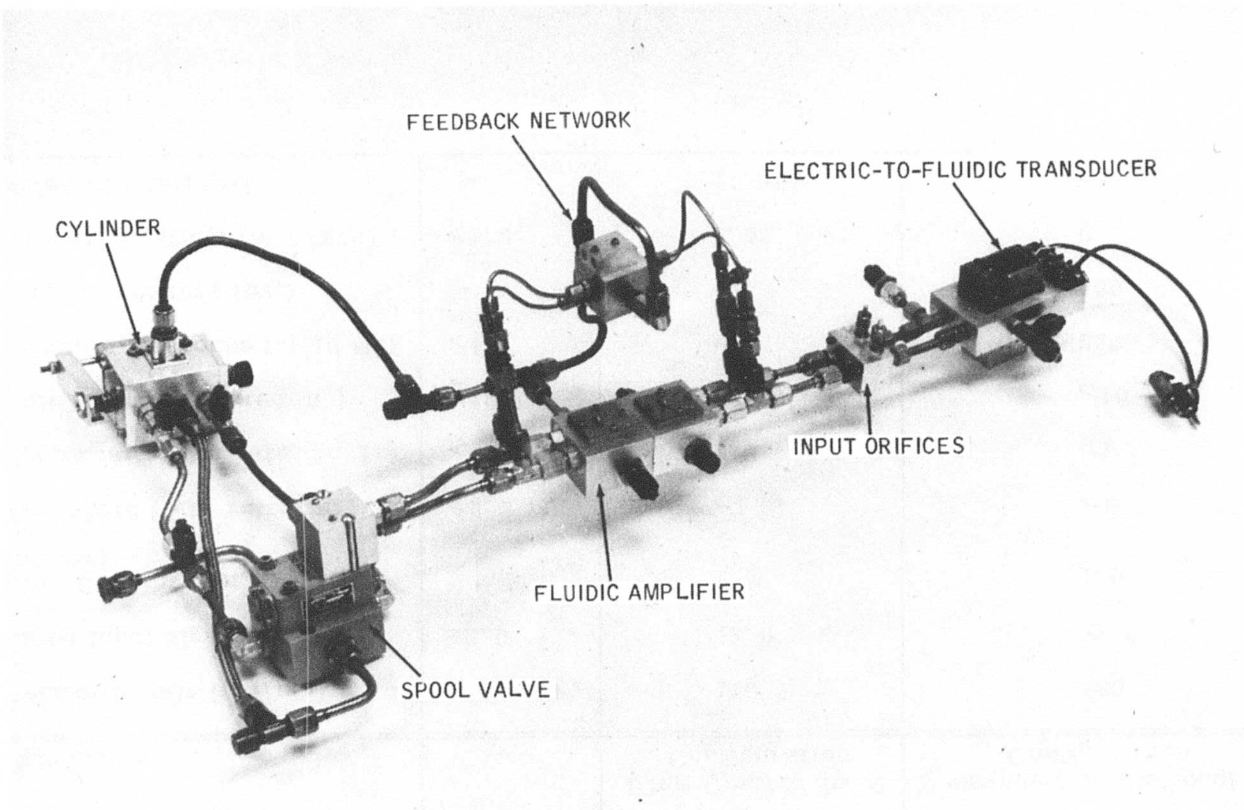


Figure 28. Servoactuator Breadboard Assembly

TABLE 2. SERVOVALVE NOMINAL PERFORMANCE

Parameter	Goal	Flow Control Spool Configuration	Pressure Control Spool Configuration
Pressure Gain (psi/psi)	100 ± 20%	110	100
Rated Input Signal (psid)	±2.0	±2.0	±2.0
Flow Capacity (in.^{3}/sec at 250 psi)	≥ 0.5	1.0	1.0
Threshold (% of rated in.)	≤2.5	<1	<1
Hysteresis (% of rated in.)	≤4	1.3	<1
Null Bias (% of rated in.)	≤10	<10	<10
Dynamic Response (at 10 Hz)	<45°	<45°	23°
Supply Pressure (psi)	—	400	400
Internal Leakage (in.^{3}/sec)	<1.0	0.53	0.53
Noise (Output) (psid)	—	±1.25	±3

TABLE 3. SERVOACTUATOR NOMINAL PERFORMANCE

Parameter	Goal	Flow Control Spool Configuration	Pressure Control Spool Configuration
Gain (in. /psid)	0. 1 + 20%	0. 11	0. 12
Linearity (% of full out)	<7. 5	7. 5	11
Actuator Stroke (in.)	±0. 2	±0. 2	±0. 2
Threshold (%)	—	1	3
Hysteresis (%)	—	2	5
Actuator Velocity (in. /sec)	≥2. 0	≈4	≈4
Null Hunting (Noise) (in.)	<0. 01	0	0
Null Bias (% of full out)	< 10	<10	<10
External Load Resistance (in. /lb)	≤0. 005	0. 0055	0. 0055
Response (-3 dB) (Hz)	>3	3. 5	>10
Internal Leakage (in.^{3}/sec)	<1. 0	0. 53	0. 53
Supply Pressure (psi)	—	400	400

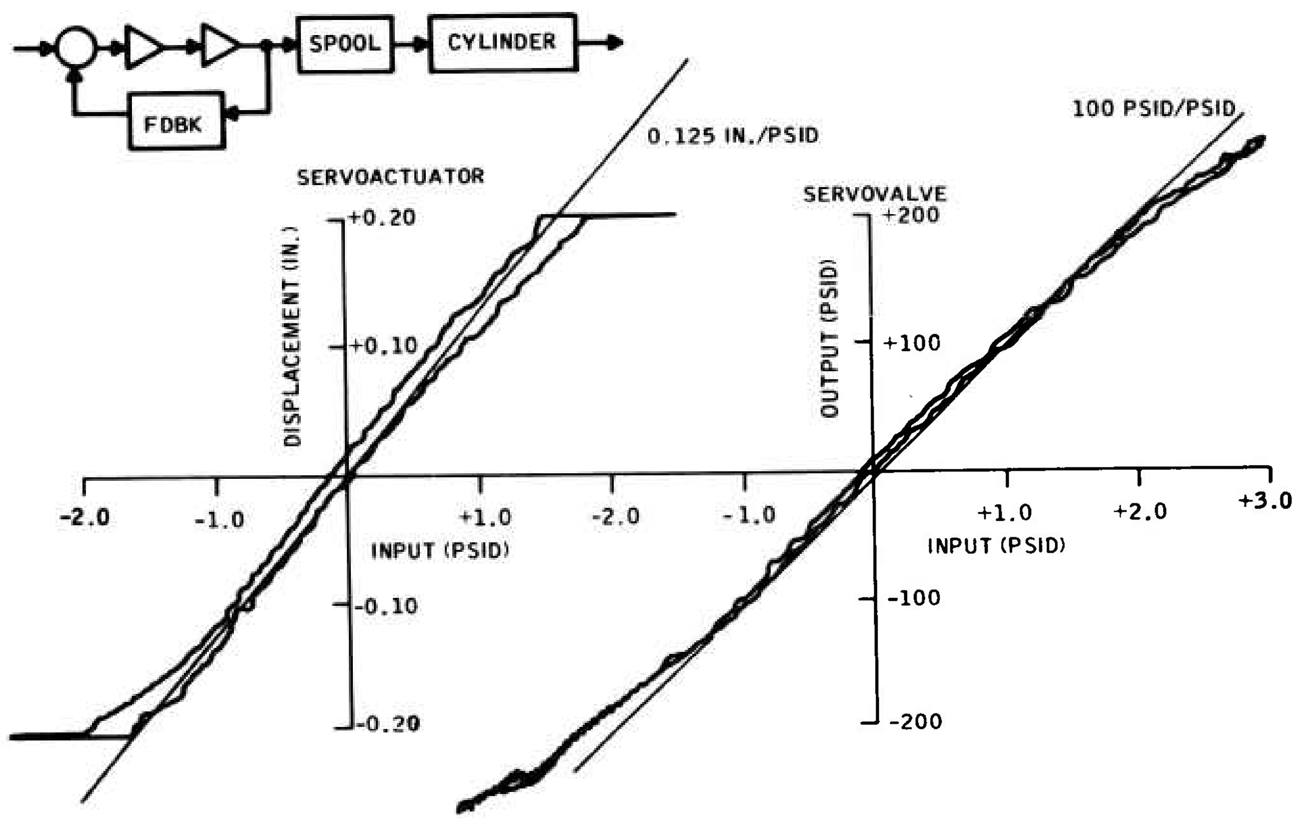


Figure 29. Servo Performance with Feedback (Pressure Control Spool)

valve) with feedback. The stepping action shown in the curves is associated with the effect described. Figure 30 shows servoactuator gain curves, for the same conditions, with full output range shown in one case and ±50 percent range shown in the other. In both cases, the deadband is ±4.5 percent of the total range. The improvement in deadband, over the cylinder only case shown in Figure 21, is attributed to the dither effect of noise in the circuit.

Figure 31 shows response for the total servoactuator. The curve is close to a second-order system with a natural frequency of 12 Hz and a damping ratio of 0.2. Figure 32 shows response from input to spool valve output for both cylinder-locked and cylinder-unlocked conditions. With the cylinder unlocked, the response is similar to the total servoactuator, except for a second higher frequency resonance associated with the valve. With the cylinder locked, the lower frequency resonance associated with the spool and cylinder is eliminated leaving the higher frequency servovalve response.

The performance of the servovalve and the servoactuator using the pressure control spool valve is also summarized in Tables 2 and 3. As was the case with the flow control spool configuration, the nonlinearity is relatively large. This is associated with both the hysteresis and the rounding of the gain curve near full stroke. Although the response of this configuration is high, the low damping ratio (see Figure 31) is an undesirable feature.

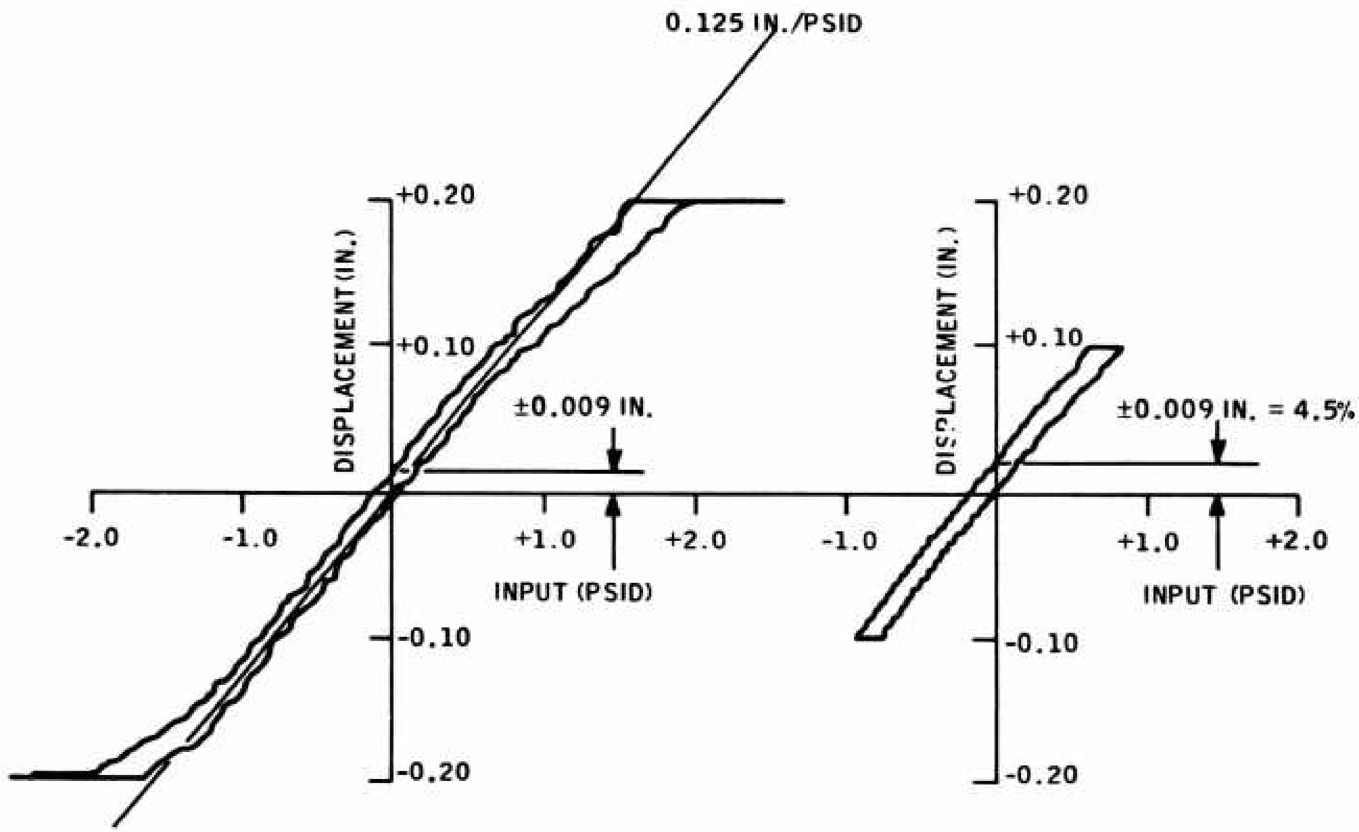


Figure 30. Servoactuator Performance (Pressure Control Spool)

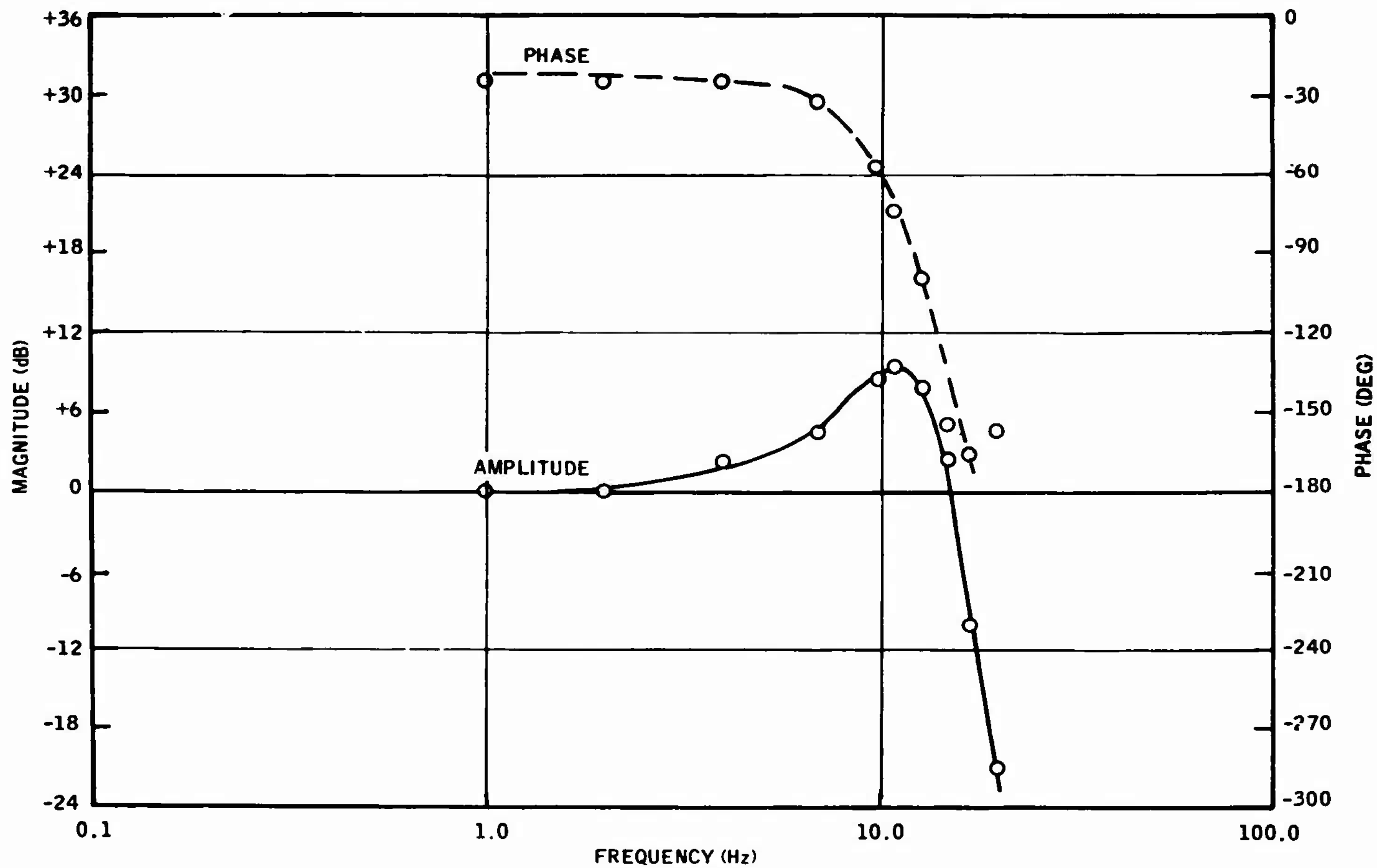


Figure 31. Servoactuator Response (Pressure Control Spool)

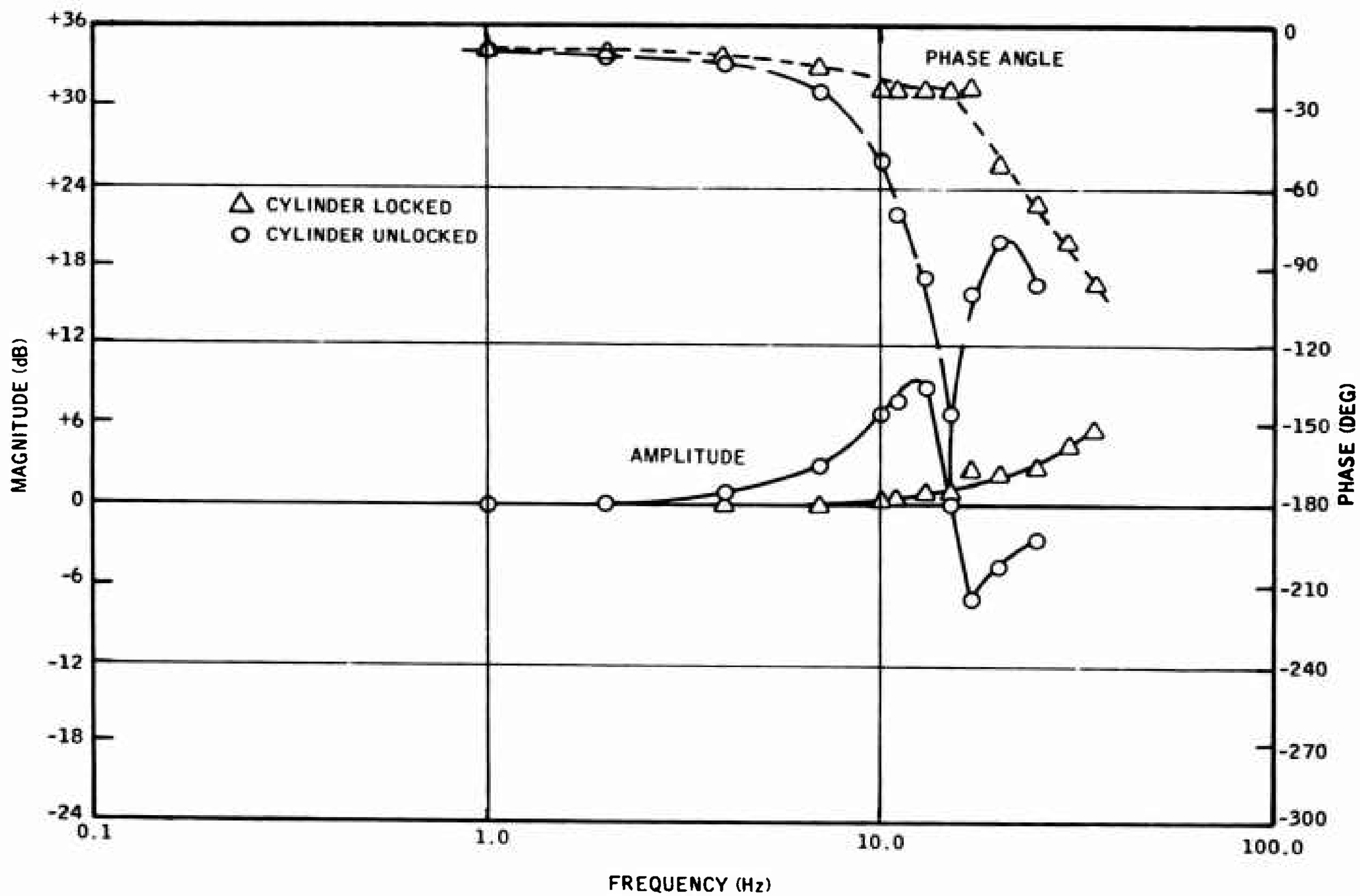


Figure 32. Servovalve Response (Pressure Control Spool)

HIGH/LOW TEMPERATURE TESTS

A problem of excessive gain change with fluid temperature variation was encountered with both servoactuator configurations. The problem was traced to gain change in the orifice feedback network, which is also the cause of the gain change with supply pressure variation shown in Figure 23. Gain of the fluidic amplifier elements decreased substantially at low oil temperature, particularly during the initial temperature tests when the FG1004AA06 amplifier design was used. The short power nozzle FG1004AA05 design improved low temperature performance; however, the gain of the amplifier cascade still decreases at low oil temperature.

Figure 33 shows servoactuator gain change over the oil temperature range of 40° to 180°F for the two spool valve configurations. The trend for the two configurations is similar. The increase in gain with a decrease in oil temperature from 180° to 80°F is associated with gain change in the feedback network. One problem is the small valves in the feedback network that are used to adjust the gain. These valves, due to their design and small metering clearances, act as viscous resistors. The result is higher feedback gain at high oil temperature where the viscosity is lower.

The decrease in gain below 80°F oil temperature is due to the decrease in amplifier gain. The drop is much greater with the pressure control spool configuration because there is less feedback around the amplifiers.

In addition to the effect of oil temperature on gain, both configurations exhibited null shifts of approximately ±100 percent of rated input over

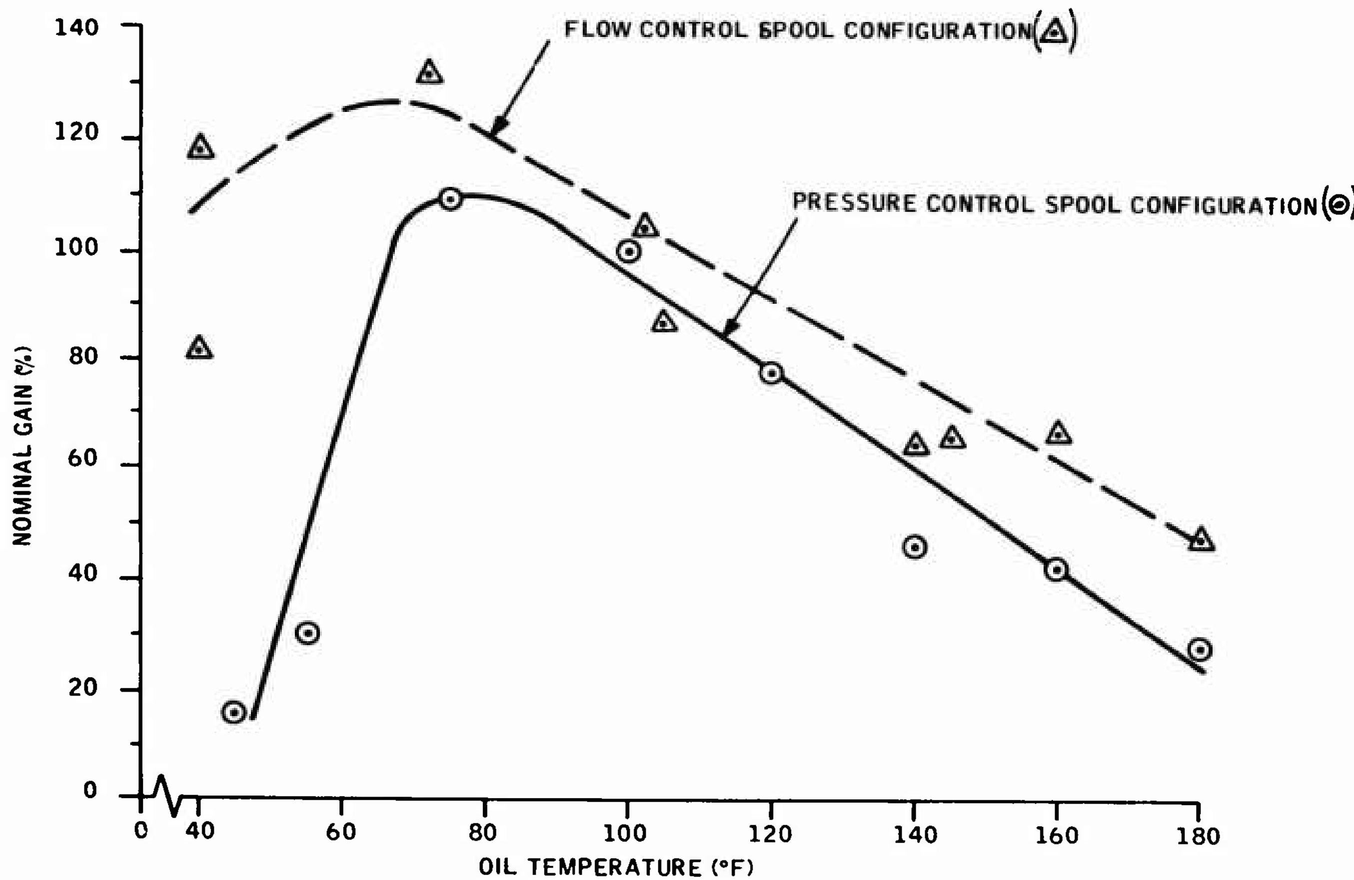


Figure 33. Servoactuator Gain Versus Oil Temperature

the temperature range. This is attributed to the input restrictors and the feedback network restrictors. It is believed that improvements in these resistor circuits would greatly improve both the gain and null bias characteristics of the servoactuator.

Higher oil temperature (greater than 120°F) also had an effect on the stability of the pressure control spool configuration. At these temperatures, the servoactuator had a tendency to oscillate at certain input signals, indicating that the low damping ratio decreases even further when the fluid viscosity is decreased. One possible solution to this problem is the addition of a lag to the servovalve. Because of the high response of this configuration, it should be possible to add additional lag to stabilize the actuator and still meet the overall response goal. Because of the instability problem at high oil temperature and the low gain at low oil temperature, it was not possible to obtain frequency response data at the temperature extremes. Response data was taken at 40° and 180°F fluid temperatures for the flow control spool configuration. At 40°F the 3 dB attenuation frequency was less than 0.5 Hz, while at 180°F fluid temperature it was approximately 2 Hz. In both cases, this is a significant reduction from the nominal performance response. Additional testing would be required to determine the cause of the response change with oil temperature.

SECTION IV
PRODUCTION COST ESTIMATE

The production cost for the servoactuator was estimated for quantities of 100, 300, and 500 units. Because the units tested on this program were breadboard models and a production design was not defined, parts of approximately equivalent complexity from an earlier fluidic SAS program were used where definition of a particular part did not exist. The estimate includes cost for fabrication of the servoactuator parts, assembly of the unit, and acceptance testing.

Table 4 is the servoactuator parts list used for the estimate. Vendor quotes were obtained for the spool valve, filter, O-rings, and screws.

TABLE 4. PARTS LIST FOR PRODUCTION COST ESTIMATE

Quantity	Part/Assembly
1	Spring-centered Cylinder Assembly
1	Spool Valve (Moog Inc.)
1	Electroformed Manifold
2	Fluidic Amplifiers
1	Feedback Valve Assembly
1	Filter Element
1	Filter Retainer
22	O-ring Seals
8	Orifice Inserts
10	Socket Head Screws (4-40)
4	Socket Head Screws (8-32)

Because the two servoactuator configurations are so similar and the prices for the two spool valves (flow control and pressure control) were essentially the same, only one cost estimate was made. It applies to either the flow control spool configuration or the pressure control spool configuration.

The resultant cost estimates, in 1978 dollars, for the three quantities are given in Table 5.

TABLE 5. PRODUCTION COST ESTIMATES

Quantity	Price ($)
100	2,200
300	1,950
500	1,800

These are the recurring costs for the unit, and they do not include the nonrecurring cost for the production design and the special tooling and test equipment that would be needed.

For comparison, a fluidic input servoactuator of the type used for the fluidic SAS systems, which have been flown on helicopters, would have a production price (in quantities of 500) of approximately $3,000. With the low-cost servoactuator design, a cost savings of approximately 40 percent is achieved over the more conventional higher performance servoactuator.

SECTION V

CONCLUSIONS

- With only minor exceptions, both servoactuator configurations meet the performance objectives at nominal supply conditions.
- Both configurations exhibit excessive gain and null variations over the 40° to 180°F fluid temperature range.
- The flow control spool configuration is judged to be the better design due to its lower temperature sensitivity and better stability.
- The estimated production cost for this servoactuator configuration is approximately 40 percent less than the cost of presently used fluidic input servoactuators ($3,000 versus $1,800).
- To obtain acceptable temperature performance, further development is required to reduce the temperature sensitivity of the orifice feedback network and the fluidic amplifier cascade.
- During the past ten years, a number of development programs have had as their objective either the improvement of the conventional fluidic input servoactuator design or the development of a new lower cost approach using more fluidic

elements. Attempts to simplify the design, even at the expense of performance, have not been completely successful. The conventional fluidic input servoactuator still provides the best combination of performance and cost for the fluidic SAS application.

APPENDIX A

SPECIFICATION FOR A LOW-COST FLUIDIC SERVOACTUATOR

SPECIFICATION NO. DS 25761-01

1. SCOPE

This specification establishes the requirements for a fluidic input, hydro-mechanical servoactuator for use in a hydrofluidic stability augmentation system. The servoactuator shall utilize a fluidic amplifier first stage driving a spool valve second stage which positions a spring centered cylinder. Simplicity and minimum cost commensurate with essential servoactuator performance is the goal. This device is a feasibility model in which the subassemblies shall be interconnected with tubing for ease of developmental testing.

1.1 Classification

The servoactuator described herein shall be classified as experimental. Performance and configuration requirements shall be design goals.

2. APPLICABLE DOCUMENTS

The following documents of the issue in effect on the date of this specification shall form a part of this specification to the extent specified herein. In the event of conflict between a referenced document and this specification, this specification shall be considered a superceding requirement.

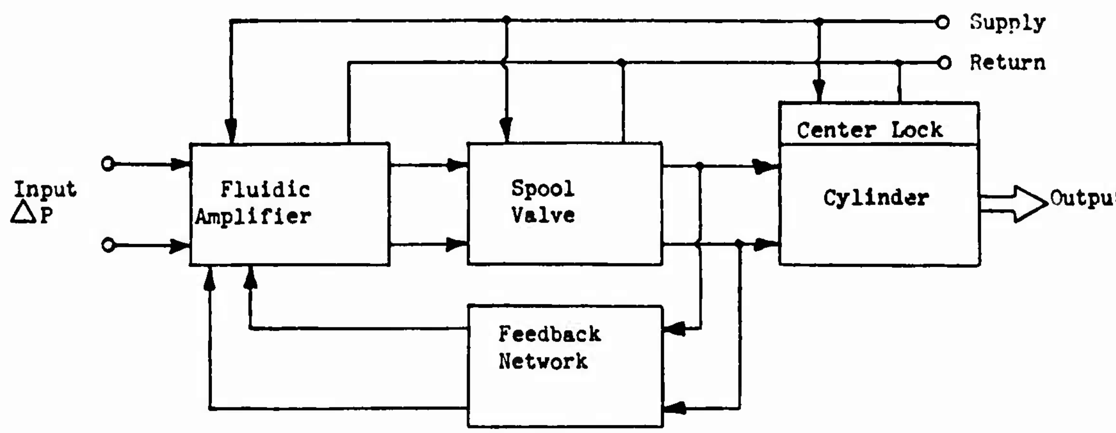
MIL-H-5440E	Hydraulic System, Aircraft Types I and II, Design, Installation and Data Requirements for
MIL-H-5606G	Hydraulic Fluid, Petroleum Base, Aircraft and Ordinance

3. REQUIREMENTS

3.1 Item Definition

The subject item shall consist of a pressure control servovalve which positions a spring centered cylinder. The servovalve first stage shall be a fluidic amplifier with the output driving the second stage spool valve. Servovalve output load pressure is fed back to the input to obtain a closed loop pressure control servovalve. The spring centered cylinder shall provide an output displacement proportional to cylinder port differential pressure. It shall also include an integral center locking mechanism which locks the cylinder in its centered position when hydraulic pressure is removed.

3.1.1 Block Diagram



3.1.2 Servoactuator Features

The servoactuator shall incorporate the following features:

a. A fluidic amplifier consisting of two stages of the FG1004AA06 design.

b. A conventional four-way spool valve. Both a spring centered flow control spool valve and a pressure control spool valve shall be evaluated for this application.

c. A MG1018AA01 spring centered cylinder with an integral center locking mechanism.

d. A pressure feedback network consisting of an orifice and valve circuit which can be adjusted to vary the amount of negative feedback from the spool valve output to the fluidic amplifier input.

e. Interconnections between the above sub-assemblies shall be made with standard tubing and fittings.

3.1.3 Hydraulic Supply

The subject equipment shall be designed to operate in a Type I (-65 to +160°F), Class 1500 psi hydraulic system conforming to MIL-H-5440.

3.2 Performance Characteristics

3.2.1 Rated Test Conditions

The servoactuator shall be tested under the following conditions unless otherwise specified.

Hydraulic Fluid:	MIL-H-5606
Supply Pressure:	250 ± 50 psig
Return Pressure:	0 - 50 psig
Input Press. Level (Null):	5 - 15 psig above return
Fluid Temperature:	100 ± 10°F
Ambient Temperature:	70 ± 5°F

3.2.2 Leakage

3.2.2.1 External Leakage

External leakage shall not exceed one drop per hour past any dynamic seal with the item stationary and normal operating pressure applied.

3.2.2.2 Internal Leakage

Internal leakage from power supply to return shall not exceed 1.0 $in.^3/sec$ at rated test conditions.

3.2.3 Servovalve Performance

3.2.3.1 Pressure Gain

The servovalve blocked load pressure gain shall be 100 psi/psi ± 10%.

3.2.3.2 Rated Input Signal

The rated input or control signal shall be ± 2.0 psid.

3.2.3.3 Flow Capacity

Servovalve flow capacity shall be at least 0.5 $in.^3/sec$ at a supply pressure of 250 psig.

3.2.3.4 Threshold

The servovalve threshold shall not exceed 0.05 psid input signal as measured from the pressure gain curve.

3.2.3.5 Hysteresis

Servovalve hysteresis shall not exceed 4 percent of rated input signal.

3.2.3.6 Null Bias

The null bias shall not exceed ±0.2 psid input signal. Null bias is defined as the input signal where the cylinder port pressures are equal.

3.2.3.7 Dynamic Response

The servovalve shall have less than 45 degrees phase lag at an input frequency of 10 Hz when operated with a blocked load.

3.2.4 Servoactuator Performance

3.2.4.1 Gain

Servoactuator position gain shall be 0.10 inches/psid ±20%.

3.2.4.2 Linearity

All test points shall fall within a ±0.015 inch band of the best straight line through the servoactuator gain curve. This requirement includes servoactuator hysteresis.

3.2.4.3 Actuator Stroke

Actuator maximum output stroke shall be ±0.20 inches ±5%.

3.2.4.4 Output Force

The actuator shall provide a stall output force of 2.0 pounds minimum at any commanded position.

3.2.4.5 Actuator Velocity

The servoactuator shall be capable of velocities of 2.0 inches/sec. minimum.

3.2.4.6 Null Hunting

The hunting (noise) of the output shaft shall not exceed 0.01 inches.

3.2.4.7 Null Bias

The actuator output shall not exceed ±0.02 inches with the input signal at null.

3.2.4.8 Null Shift with Temperature

Actuator null shift due to fluid temperature, over the range from 40°F to 180°F, shall not exceed ±0.03 inch.

3.2.4.9 Null Shift with Supply Pressure

Actuator null shift due to supply pressure change of ±20% shall not exceed ±0.01 inch.

3.2.4.10 External Load Resistance

Actuator output motion, when subjected to an external force not exceeding the actuator full signal output force, shall be less than ±0.010 inch.

3.2.4.11 Dynamic Response

The servoactuator no load amplitude response shall have a -3db bandwidth of at least 3 Hz.

3.2.4.12 Centering and Locking

The actuator piston shall center and lock when hydraulic supply pressure falls below 50 psi.

3.2.5 Environment Requirements

3.2.5.1 Temperature

The servoactuator shall meet the requirements of Par. 3.2.4 over the fluid temperature range of 40°F to 180°F.

3.3 Physical Characteristics

3.3.1 Envelope

The feasibility model servoactuator will be physically configured for ease of bench testing and will not be an integrated package.

3.3.2 Proof Pressure

The servoactuator shall withstand a proof pressure of 900 psi without evidence of external leakage or permanent performance degradation.

APPENDIX B

HISTORY OF FLUIDIC SERVOACTUATOR DEVELOPMENT

During the past 10 years, a number of programs have included or have been specifically devoted to the development of fluidic servoactuators for flight control applications. The earliest units were modifications of conventional servoactuators. The modifications included the use of bellows to convert fluidic pressure signals to force on the armature of the flapper-nozzle valve and the use of mechanical spring feedback from the output shaft to the valve first stage. One of the earliest examples of this was the servoactuator for a fluidic yaw damper system that was flown on an F101B aircraft in 1966. In 1968, a hydraulic fluidic SAS that used a similar modified fluidic input servoactuator was flight tested on a UH-1 helicopter. Both of these units were supplied by Hydraulic Research and Manufacturing Co.

During the past 10 years, over 50 servoactuators of this type configuration have been built and used on various fluidic SAS programs with generally good results. However, during this same period, a number of development programs have had as their objective either the improvement of the standard fluidic input design or the development of a new, lower cost approach using more fluidic elements. Following is a summary of these programs, the design approaches taken, and the problems that were encountered. Table B-1 presents a short summary of this information.

The first program in 1970 was performed by General Electric Co. It included the modification of conventional Hydraulic Research fluidic input servoactuators to replace the second-stage spool valve with

TABLE B-1. SUMMARY OF FLUIDIC SERVOACTUATOR DEVELOPMENT

Title	Date	Report No.	Mechanization	Program Results	Problems Encountered
Fluidic Servoactuators for Three-Axis Fluidic Stability Augmentation System	Sept. 1970	USAAVLABS Tech. Report 70-52	• Bellows input • Flapper-nozzle 1st stage • Vortex valve 2nd stage • Mechanical feedback to flapper	3 units were built, tested, and flight tested with a fluidic SAS on a UH-1 helicopter.	• Generally acceptable • Somewhat lower response and stiffness than conventional mechanization
Hydrofluidic Servo-actuator Development (Phase I)	May 1973	USAAMRDL Tech. Report 73-12	• Fluidic amplifier Input • Fluidic amplifier 1st stage • Spool valve 2nd stage • Fluidic position feedback transducer	A development model was built and tested	• High output noise • Gain variation with fluid temperature change
Advanced Hydrofluidic Servovalve	Dec. 1974	Honeywell Doc. No. W0562-FR	Conventional mechanization (bellows input, flapper-nozzle, spool valve and mech. feedback) with following changes: • 1st stage magnetic compensation • Mech. feedback from spool to flapper (2 feedbacks)	A development model was built and tested. Results showed that servo response and threshold is limited by the available fluidic input signal and the output impedance of the driving fluidic amplifier	Acceptable performance
Hydrofluidic Servovalve Development	April 1975	USAAMRDL Tech. Report 75-21	• Bellows input • Poppet valve • Mechanical feedback to force summing lever	A development model was built and limited testing performed. Problems in adjustment of the poppets prevented obtaining test data.	Adjustment of poppets very difficult with this configuration. Operational unit not achieved
Hydrofluidic Servo-actuator Development (Phase II)	Sept. 1976	USAAMRDL Tech. Report 76-25	• Fluidic amplifier input • Fluidic amplifier 1st stage • Spool valve 2nd stage • Fluidic position feedback	This program was a follow-on to the May 1973 program above. A development model was built and tested	• Slightly lower response • Gain change with fluid temperature
Investigation of a Low-Cost Servoactuator for HYSAS	June 1978	This Report	• Fluidic amplifier input • Fluidic amplifier 1st stage • Spool valve 2nd stage • Spring-centered cylinder	A breadboard model was assembled and tested. Two spool valve types were evaluated	• Gain and null change with fluid temperature • Lower response and stiffness

fluidic vortex valves. The performance was relatively close to that obtained with the conventional mechanization and was considered to be acceptable for fluidic SAS use. From a complexity standpoint, the mechanization is not substantially better than the conventional design although it eliminates the moving part spool. The trade-off is slightly better reliability for slightly decreased performance.

The next program, conducted in 1973, replaced the bellows input and flapper-nozzle first stage with fluidic amplifiers and used fluidic feedback from a fluidic position transducer. A second phase of this program, conducted in 1976, improved the amplifier design to reduce the number of amplifiers required and to reduce noise. Nominal performance was satisfactory; however, gain change with fluid temperature variation was a problem. This problem was traced to viscous resistance in the position feedback transducer. From a complexity standpoint, the spool valve and cylinder are the same as those used in the conventional design, and the position transducer (a flapper-nozzle design) is approximately equivalent in complexity to the first-stage valve in the conventional servoactuator. Therefore, the difference is basically fluidic amplifier input versus bellows input. The fluidic amplifier input eliminates the moving part bellows, but it requires that the pressure levels at the SAS output and the servoactuator input be matched for proper operation.

The advanced hydrofluidic servovalve program conducted in 1974 was to incorporate a number of improvements in the conventional fluidic input servoactuator with the objective of improving reliability and performance. The modifications include:

- Position feedback from the spool valve as well as from the actuator.

- Electrodeposited nickel bellows force capsules for improved performance and size.

- Porting isolation valves at the servovalve interface to prevent air insertion during installation.

- Magnetic compensation to reduce the first stage stiffness-to-ground and thus increase the first-stage flow and pressure gain and improve overall servovalve response.

The modified design provided somewhat improved performance; however, it showed that the basic physical restraint on servoactuator response and threshold is the available fluidic input signal and the output impedance of the driving fluidic amplifier. From a complexity standpoint, the modified design was at least as complex as the conventional mechanization.

The 1975 hydrofluidic servovalve program attempted to simplify the fluidic input servovalve design by replacing both the flapper-nozzle and spool valves with a single-stage poppet-type valve. Problems were encountered in the adjustment of the poppets for proper operation, which prevented obtaining operational data on this mechanization.

The final design approach is the one covered by this report. The initial intent was to replace both the flapper-nozzle first stage and the spool second stage with a single fluidic amplifier cascade, and to replace the mechanical feedback actuator with a spring-centered

actuator. Initial tests showed that this approach would not provide the desired dynamic response without using excessive flow. Therefore, the second-stage spool valve was retained in the final mechanization. This approach has lower response and considerably lower stiffness than the conventional approach, and has a temperature sensitivity problem that must be resolved before use in actual fluidic SAS application. From a complexity standpoint, this approach eliminates both the bellows input and the more complex position feedback at the expense of reduced performance.

LIST OF SYMBOLS

A_p	cylinder area ($in.^2$)
F_p	pressure-generated cylinder force (lb)
F_s	spring-generated cylinder force (lb)
K_a	fluidic amplifier pressure gain (psi/psi)
K_{fb}	pressure feedback gain (psi/psi)
K_1	cylinder leakage ($in.^3$/sec/psi)
K_s	effective centering spring rate (lb/in.)
K_s'	spring rate for each centering spring (lb/in.)
K_v	spool valve pressure gain (psi/psi)
K_v'	spool valve flow gain ($in.^3$/sec/psi)
K_2	zero load pressure droop (psi/$in.^3$/sec)
R_s	spool valve output resistance (psi/$in.^3$/sec)
S.F.	servoactuator scale factor (in./psi)
T	servoactuator time constant (sec)
X_i	control linkage displacement (in.)
ΔP_c	input to servoactuator (psi)
ΔP_o	fluidic amplifier output pressure (psi)
ΔQ_o	fluidic amplifier output flow ($in.^3$/sec)
ΔX	SAS actuator displacement (in.)
τ	droop time constant (sec)

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